# Power and The Engineer

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IN Europe conservation long ago ceased to be a fad. On the Continent conservation is a vital issue; the people are in dead earnest,—they have to be. Farms are cultivated to the very edges of the highways and railroad rights of way. The unending struggle is to make the ground yield more and to make that which is yielded go farther. The people there practise economy to a degree which is as yet undreamed of in America.

What applies to the food products applies also to fuel. Industrial prosperity demands a fuel of reasonable cost.

Some countries, as for example, Sweden, Norway, Denmark, Finland and Holland, have practically no coal deposits and are forced to import what they need. During war times a country would be seriously handicapped if its fuel supply were suddenly interrupted. For this reason the governments of those countries which have little coal are keenly interested in the development of a suitable substitute which can be produced within their own borders.

Peat, which is found in great abundance in all parts of Europe, has given satisfaction as a fuel. In Holland it has been used for centuries. The production of peat at the present time is over a million tons per year. In Russia the annual output amounts to over four million tons. The peat-fuel resources in other parts of Europe are proportionately well developed. America also has vast peat deposits. The United States, exclusive of Alaska, has over eleven thousand square miles of peat bogs. Canada has more than thirty-seven thousand square miles of bog land.

In many places, especially in Canada, peat fuel would find a very ready sale if it were offered at a price which would compare favorably with that of coal.

It is easy to burn and is comparatively free from smoke. Its heat value is about 40 to 60 per cent. of that of coal, and it would appear that the most efficient way in which to use peat for power generation is in a gas-producer plant.

The peat-fuel industry has not been a success in America due to the comparatively low cost of coal and the imperfect machinery which has been used for handling the peat.

The Canadian government is doing a great work at its experimental stations in demonstrating that peat can be used profitably for industrial purposes. These stations are described in this issue.

The United States Government also is making considerable effort to encourage the development of the possibilities which peat fuel possesses. Several large industrial and engineering concerns are making investigations to determine for their own benefit the availability of peat for power generation.

These facts indicate that the time is not far distant when peat will play an active part in the conservation of the American coal supply just as it is doing at the present time in the less richly endowed countries of Europe.

The methods of handling and using this product are not as yet highly perfected. However, the fact that so much interest is being taken in peat, is a token that great progress may be expected within a few years.

# Canada's Experimental Peat Plants

By A. R. Maujer

*At the government bog lands peat fuel is dug by hand, pulped in a mixing machine and spread out on the ground to dry. The total labor cost of all of the operations, including the loading of the peat into freight cars, is $1.63 per ton. At the fuel-testing station gas is made from the peat and is used in a Koerting gas engine which drives a 50-kilowatt generator.*

The coal deposits of Canada are all located in the western and eastern portions of the country; practically no coal is found in the central parts. The anthracite which is used in the central provinces is obtained from the Pennsylvania district. Due to the cost of transporting it such a great distance the cost is high. In Ontario and Quebec anthracite of very ordinary grade costs $7.50 per ton; in Manitoba the cost is as high as $10. For these reasons a satisfactory substitute for coal which could be made to compete with it commercially would find a ready market.

Peat, which is found in great abundance through Canada and the United States, is just such a substitute. Doctor Chalmers, of the Geological Survey of Canada, has estimated that there are considerably more than 37,000 square miles of workable peat-bog land in Canada. The depth of the bogs varies from 5 to 10 feet. This estimate is based on investigations which have approached thoroughness only in the territory east of Lake Superior. Throughout the western section of the country vast areas exist which have not been investigated but which undoubtedly have immense peat deposits.

Numerous attempts to manufacture fuel from peat have been made from time to time in various parts of Canada. None has proved entirely successful. The long succession of failures has served to discourage further efforts. The Canadian Government, recognizing the economic possibilities in a thriving peat-fuel industry, is endeavoring to stimulate the interest and enterprise of bog owners and manufacturers by demonstrating that peat fuel can be produced cheaply and that power for industrial purposes as well as heat for domestic use can be obtained therefrom. To accomplish this a portion, comprising about 300 acres, of what is known as the Alfred bog has been acquired and fuel is being manufactured. The peat is used at the fuel-testing station in a peat producer-gas plant of 60 horsepower capacity. The bog is situated about 40 miles east of Ottawa in Prescott county, province of Ontario. It covers an area of approximately 6800 acres. The peat varies in depth from 3 to 17 feet.

That the peat is suitable for fuel is shown by the following analysis which is given in Bulletin No. 4 of the Canadian Department of Mines.

ANALYSIS OF PEAT.
Absolutely Dry.

| | Per Cent. |
|---|---|
| Volatile matter | 68.23 |
| Fixed carbon | 26.00 |
| Ash | 5.77 |
| Nitrogen | 1.76 |
| Sulphur | 0.218 |
| Phosphorus | 0.033 |
| Calorific value in B.t.u. | 9005.000 |

Before actual manufacturing operations were started at the Government bog about 24,000 linear feet of drainage ditches were run late in the summer of 1909. This required the excavation of about 9300 cubic yards. The excavating was done entirely by hand, and cost approximately 8½ cents per cubic yard.

While the ditches were being dug an Anrep peat machine, together with other machinery and equipment, was purchased, brought to the Government bog and made ready for operations which were started in the summer of the present year. The machine has a capacity of about 30 tons per day.

The manner in which the peat is dug and handled is shown in Fig. 1. The men work at different levels so as to secure uniformity in the composition of the peat after it has passed through the machine. The peat is dug out with large sharp-edged spades and thrown onto the elevator. At the upper end the peat falls off into the hopper of the mixing machine in which all roots and stems are cut into small pieces and the peat is so thoroughly mixed that it resembles a pulp when it emerges. The cutting is accomplished by a number of knives mounted on a shaft which revolves at a speed of about 260 revolutions per minute. These knives pass sets of stationary knives which are fixed to the interior surface of the casing. A number of broad helix-shape blades mounted on the same shaft, do the mixing and force the peat out of the mouth of the machine onto a belt conveyer. The peat falls from the end of this conveyer into small steel dump cars which are hauled to and from the drying field by an endless cable. The cars travel on

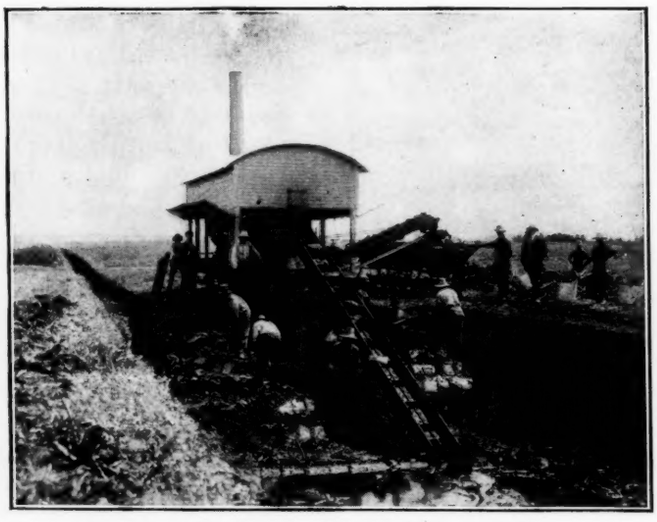

FIG. 1. THE MIXING MACHINE AND METHOD OF DIGGING THE PEAT

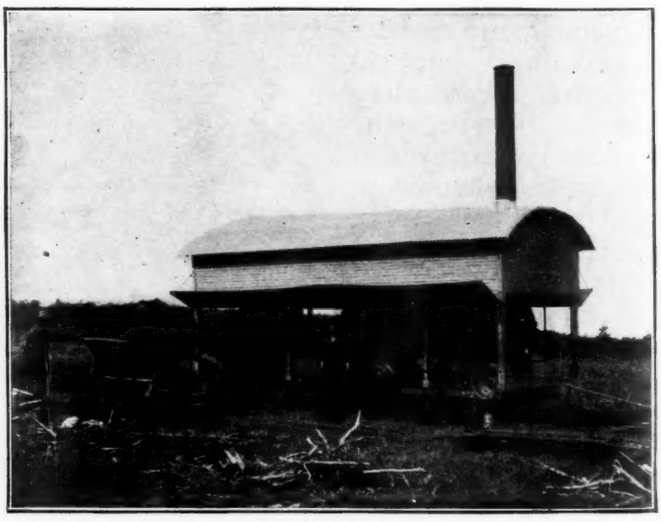

FIG. 2. ANOTHER VIEW OF THE MIXING MACHINE

light portable steel tracks which are laid down on the roughly leveled surface of the bog.

The drums which move the cable may be seen in Fig. 2. The manner in which the cable is gripped by the car is also shown in this figure. The upper of a pair of jaws attached to the truck of the car is raised and lowered by a cam and weighted lever. When the upper jaw is raised the cable rides on a grooved roller mounted on a spindle in the middle of the lower jaw and does not come in contact with either jaw. When the lever is thrown over the weight holds the movable jaw down and the cable is firmly gripped. The boiler and engine are built integral with each other and are made as light and compact as is consistent with durability. The rated capacity of the engine is 35 horsepower. The boiler is fitted with a grate suitable for burning either peat or brushwood. The entire outfit is mounted on a steel frame which is carried on car wheels. When it is necessary to move the apparatus, it is warped along over ordinary steel rails by means of a cable which runs through a pulley attached to an anchor and is wound over a slowly moving geared drum on the machine.

When a car full of peat arrives at the drying field, one man disconnects the grip and dumps the peat from the car into a Jakobson field press which travels along at the side of the tracks. The press is shown in Fig. 3. It consists of a wooden frame the front part of which travels on a wide wooden roller. The peat is roughly spread out on the ground within this frame by two men. The rear part of the frame is covered. Under the cover there is another wide wooden roller. This roller is set about 5 inches above the surface of the ground and is rotated by the front roller to which it is connected by chains located at the sides of the frame. The roller and frame press the peat out into a smooth flat strip about 6 feet wide and 5 inches deep. The back of the frame carries a set of 14 wooden knife blades which are held down by as many weights. The knives divide the strip of peat into 15 ribbons of equal width. These are cut crosswise into lengths of about 12 inches with suitable circular knives by the man who empties the cars. The object of cutting the peat up in this way is to hasten the drying by exposing more surface to the air. The press is slowly dragged along at the desired rate by a cable wound over a drum at the engine.

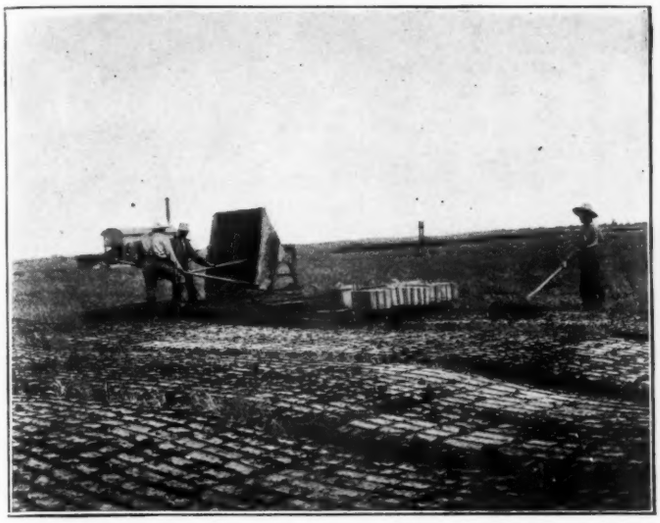

FIG. 3. THE JAKOBSON FIELD PRESS

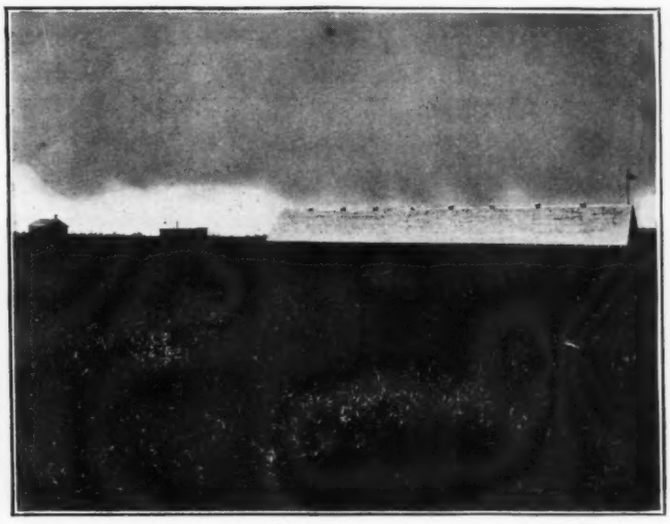

FIG. 4. PEAT-STORAGE SHED, BLACKSMITH SHOP AND OFFICE

Fig. 5 is a diagrammatic plan of the

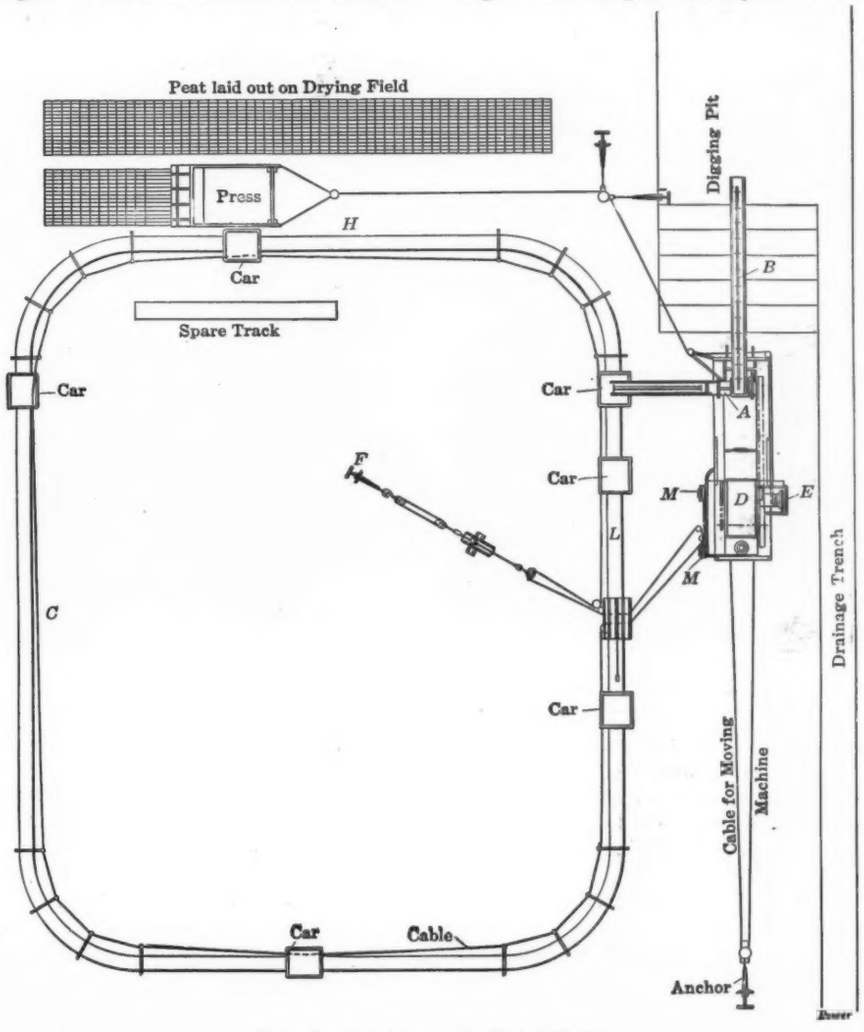


FIG. 5. DIAGRAM OF PEAT FIELD

field. The engine and boiler are indicated at *D*, the mixing machine is at *A* and the elevator at *B*. The cable for hauling the press is operated by the drum *E* and the car cable by drums *M*. When it is necessary to move the tracks *H* to start a new strip of peat, a section is taken out of the sides *C* and *L*. The slack in the car cable is taken up by shifting the anchor and sheave *F* the required distance.

The peat is allowed to lie undisturbed for seven to ten days or longer, depending upon climatic conditions, until it is firm enough to be turned. After it is turned the peat is allowed to remain on the drying field for about ten days longer. It is then stacked. Each slab is spaced an inch or two away from its neighbors so that air may circulate through the stack and the drying process continue. The peat which is to be carried in stock for any length of time is removed from the stacks after about 30 days and placed in the storage shed which is shown in Fig. 4. The storage shed is of cheap, wood construction, 160 feet long, 22 feet wide and 18 feet high. The first small house beyond the shed is the blacksmith's shop; the second is the office.

Thirteen men and three boys are required to dig, machine and spread the peat. The men are paid $1.75 per day and the boys $1. This makes a total cost of $25.75 per day to lay the peat down on the drying field. At the rate of 30 tons per day the cost per ton is about 86 cents. The cost of turning the peat is 7 cents per ton and the cost of placing it in stacks is 25 cents. It costs 20 cents per ton to place the peat in the storage shed and 25 cents to load it from here into freight cars. This makes a total cost of $1.63 per ton for digging, spreading, handling and loading the peat for transportation. To this must be added the freight charges, the interest on the investment and the amortization. These items cannot be estimated from the data at present available. It is believed, however, that these will not cause the cost of the peat to become so great as to eliminate it from competition with coal at $7 or more per ton.

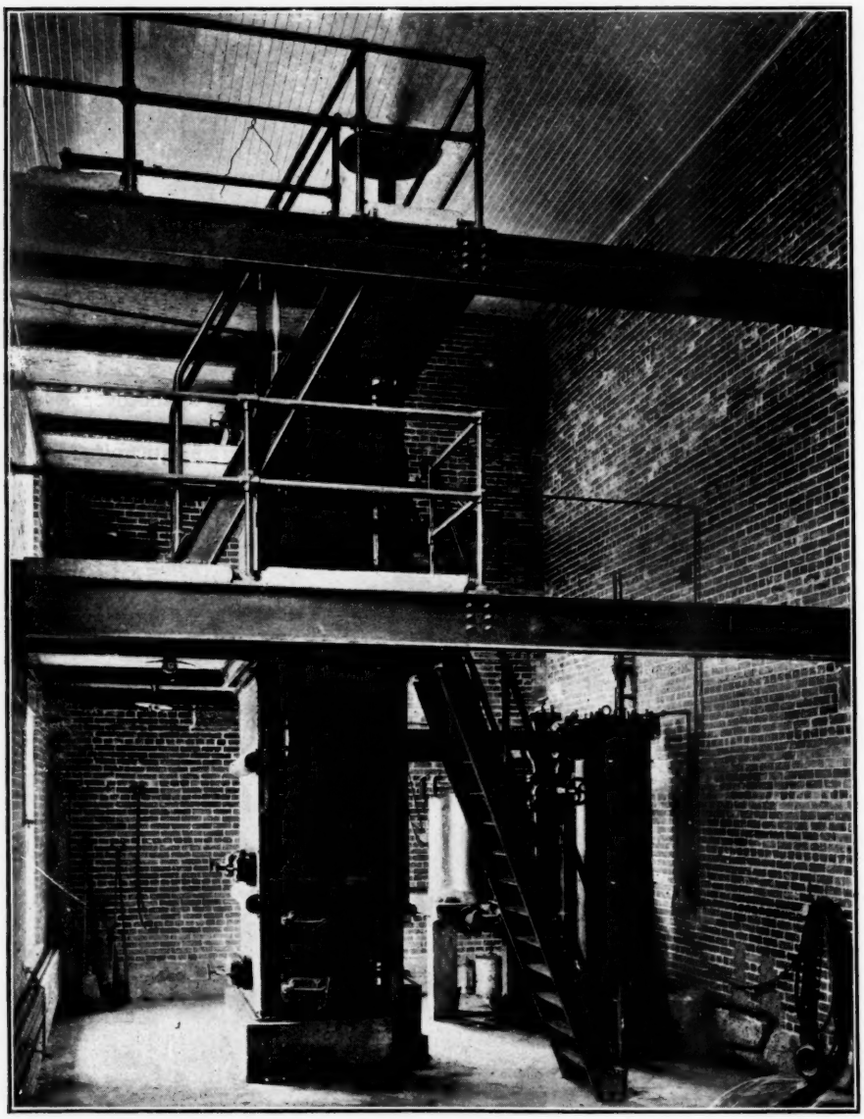

FIG. 6. THE KORTING GAS PRODUCER AND GAS-CLEANING APPARATUS

The Canadian Government will keep all of the peat that it requires for testing purposes and sell the rest at a suitable price.

### FUEL-TESTING STATION

The Government fuel-testing station is located in Ottawa on Dolly Varden and Division streets. It was erected during the summer of 1909. At present the plant is equipped for testing peat fuel only. The apparatus consists of a Körting double-zone suction gas producer, the necessary purifying apparatus, a Körting gas engine and a 50-kilowatt Westinghouse direct-current generator. The producer room is large enough to hold additional apparatus and producers of other types are to be installed in the near future. These producers will be especially designed for bituminous coals and lignites. Part of the producer room and the Körting producer are shown in Fig. 6. The producer is 15 feet high, 2 feet 9 inches wide and 5 feet long.

The gas-cleansing apparatus is shown at the right of the producer. This consists of a wet tar separator and a coke scrubber.

The engine and generator are shown in Fig. 7. The engine is a single-cylinder, four-stroke-cycle machine, fitted with a throttling governor. The diameter of the cylinder is 15 inches, the stroke is 24 inches. The speed is 200 revolutions per minute. The air supply for the engine is obtained from the interior of the engine room through a brick duct built against the wall. This duct may be seen in Fig. 7 to the right of the engine. With this arrangement the temperature of the air supply remains practically constant throughout the year. During tests the current generated by the dynamo is absorbed by either the bank of five hundred 16-candlepower incandescent lamps, which is located on the end wall of the room, or by the iron grid which may be seen against the wall at the left of the view.

When the producer is started up, suction is furnished by a small centrifugal exhauster belted to an alternating-current motor. Current for the motor is obtained from the city-lighting company's mains. The motor and the exhauster may be seen at the extreme right of the view. The engine is started with compressed air.

The gas-testing apparatus is located on the right wall of the engine room. A detail view of this is given in Fig. 8. Gas is drawn from the system by a small Root blower and passed through a water-pressure regulator to a Junker gas meter. From the meter the gas goes to the calorimeter which is also a Junker instrument. The average effective heat value of the gas is about 126 B.t.u. per cubic foot. The producer requires from two and one-half to three pounds of peat per

brake horsepower per hour. The plant has been in actual operation for such a short time that no specific data in regard to the cost of operation and the performance of the apparatus are available at this writing.

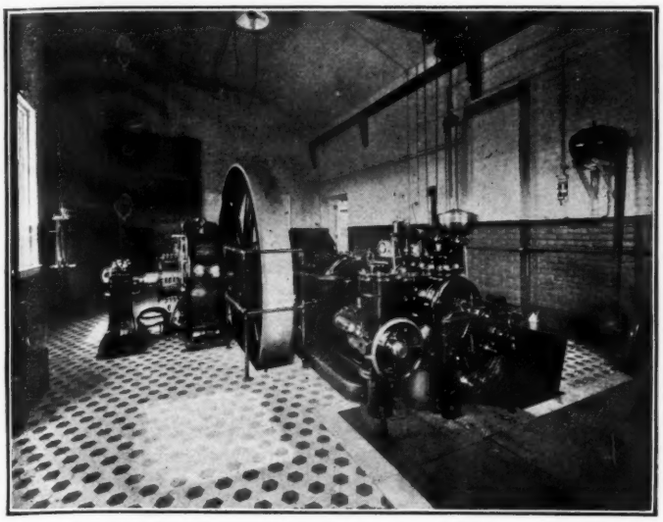

FIG. 7. A VIEW OF THE ENGINE ROOM

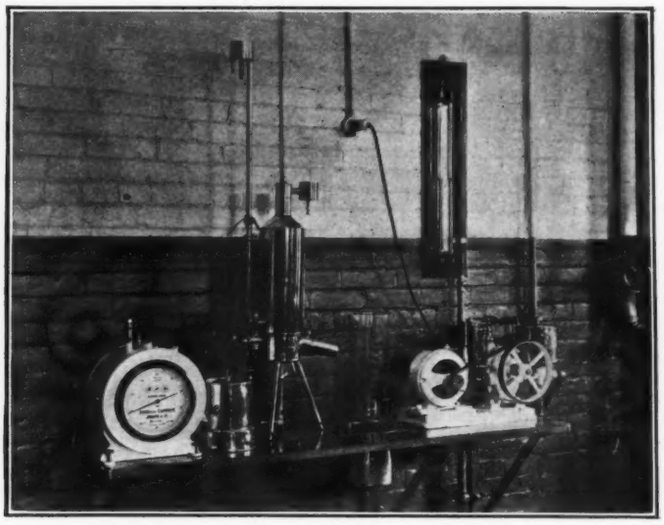

FIG. 8. GAS-TESTING APPARATUS

We acknowledge our thanks to Dr. Eugene Haanel, Canadian Director of Mines, for permission to obtain the material for this article. Much of the data relative to the peat-fuel manufacturing plant of Alfred was supplied by Aleph Anrep, peat expert for the Canadian government. Information relative to the fuel-testing station was secured through the courtesy of B. F. Haanel, chief engineer of the fuel-testing division of the Department of Mines.

# Notes on Economizer Practice

## By George H. Gibson

*A discussion of the relations between boiler and economizer heating surfaces and flue-gas temperatures, supplemented by economic data upon economizer practice with various operating conditions and different costs of coal.*

Ordinarily 20 to 50 per cent. of the heat value of the fuel burned under a steam boiler escapes to the atmosphere in the form of sensible heat of the gases passing up the chimney. The percentage of the total heat of the fuel carried away by the flue gases at various temperatures, assuming that 26 pounds of air are used to burn a pound of coal containing 14,500 B.t.u., is shown in Table 1.

Each heat unit utilized in a boiler represents a certain cost in fuel and boiler upkeep. This affords a basis upon which can be estimated the value of the heat recovered by any heat-saving device, such as an economizer. The cost of obtaining heat in a boiler is made up of the following items:

Cost of fuel.

Cost of handling fuel and ashes.

Repairs and upkeep of the plant.

Fixed charges against the boiler.

The average first cost of the boiler and setting may be taken as $15 per boiler horsepower, and of the stoker, grate and draft apparatus as $5 per horsepower. The annual fixed charge is represented by the interest on this investment, plus the amount that must be laid aside yearly to provide for the indefinite continuance of the plant; that is, sinking fund and maintenance. Placing the interest at 5 per cent., and the sinking fund and maintenance charges at 12 per cent. of the first cost represents a total annual expense, exclusive of fuel, of $3.40 per boiler horsepower. The annual cost of fuel varies according to the amount of hours the plant is operated daily and to the number of days per year. The costs for incidental labor and power around the boiler plant are small, compared with the other factors, hence they may be assumed as constant without introducing any serious error.

With coal at $3 per ton, containing 14,500 heat units per pound, and the boiler operated at full load 10 hours per day and 310 days per year, at an average efficiency of 65 per cent., the annual cost for each boiler horsepower will be $16.28 for fuel and $3.40 for fixed charges, making a total of $19.68 for 3100 horsepower-hours. With coal at $3 per ton, and operating 24 hours per day and 365 days per year, the other factors remaining the same, the annual cost of 8760 boiler horsepower-hours becomes $49.40. Both figures, especially the latter, are below the actual costs incurred in practice as boilers are seldom operated with a load factor of 100 per cent. on all-day service, and it is, in general, necessary to have surplus boiler capacity in order to provide for cleaning and repairs; this would increase the fixed charges per boiler horsepower developed.

TABLE I. PERCENTAGE OF HEAT ESCAPING IH CHIMNEY GASES AT VARIOUS FLUE TEMPERATURES, ASSUMING 26 POUNDS OF AIR PER POUND OF COAL AND 14,500 B.t.u. PER POUND OF COAL.

| Chimney Temperature, Degrees Fahrenheit. | B.t.u. in Gases per Pound of Coal. | Percentage of Total Heat in Gases. | Ratio of Heat in Flue Gases above 60 Degrees to Heat Absorbed by Boiler, Assuming 15 per Cent. Loss Through Grates and Radiation, per cent. |
|---|---|---|---|
| 200 | 905 | 6.25 | 8 |
| 300 | 1555 | 10.7 | 14.4 |
| 400 | 2200 | 15.2 | 21.8 |
| 500 | 2850 | 19.68 | 30.2 |
| 600 | 3400 | 23.5 | 38.3 |
| 700 | 4150 | 28.6 | 50.6 |
| 800 | 4790 | 33.0 | 63.4 |
| 900 | 5440 | 37.5 | 79.0 |
| 1000 | 6090 | 42.0 | 97.5 |

The transmission of heat by convection from hot gases to iron surfaces in contact with water has been carefully investigated, and although affected by such factors as the velocity of the gases and the condition of the surfaces, it may be stated that for such temperatures as are found at the stack end of a boiler and in an economizer, the coefficient of transmission varies from 2½ to 4½ B.t.u. per square feet of surface per hour, per degree difference of temperature between the hot gases and the water.

One boiler horsepower-hour is equivalent to 33,300 B.t.u. Taking the lower value .2½ B.t.u. as the coefficient of transmission and letting $D$ represent the difference in temperature between the gases and the water at the given point, each square foot of additional heating surface would recover

$$\frac{D \times 2\frac{1}{2}}{33{,}300} = 0.000075\ D$$

horsepower. Assuming that the boilers are rated on the basis of 10 square feet of heating surface per horsepower, the annual cost of maintenance for a square foot of heating surface would be $0.34; and the annual cost of a boiler horsepower recovered by the additional surface would be

$$\frac{0.34}{0.000075\ D} = \frac{4530}{D}\ dollars$$

Equating this to the original cost of a boiler horsepower-year of 310 days at 10 hours each, namely, $19.68, $D$ is found to be 231 degrees Fahrenheit; that is, if the difference in temperature between the gases and the water is more than 231 degrees Fahrenheit, the additional surface is more than paying for itself by the heat that it is recovering from the gases; if the difference in temperature is less, the additional surface is not paying for itself. This determines at once how far boiler surface can be extended profitably. Supposing the steam pressure to be 150 pounds gage, corresponding to a temperature throughout the boiler of 366 degrees Fahrenheit, the lowest temperature to which it will pay to reduce the gases by means of boiler surface is

$$366 + 231 = 597$$

degrees Fahrenheit.

In a similar manner the temperature to which it pays to reduce flue temperature may be determined. The lowest temperature to which gases should be reduced by boiler surface for various prices of coal are shown in Table 2.

TABLE 2. MINIMUM ECONOMIC TEMPERATURE OF GASES LEAVING THE BOILER, OPERATING 3100 HOURS PER YEAR.

| Price of Coal per Ton. | Cost of Year. | Critical Temperature Difference, Degrees Fahrenheit. | Economical temperature of gases leaving boiler, pressure = 150 pounds gage. Degrees Fahrenheit. |
|---|---|---|---|
| $2.00 | $14.25 | 319 | 685 |
| 2.50 | 16.97 | 268 | 634 |
| 3.00 | 19.68 | 231 | 597 |
| 3.50 | 22.40 | 202 | 568 |
| 4.00 | 25.10 | 181 | 547 |
| 4.50 | 27.80 | 163 | 529 |
| 5.00 | 30.50 | 149 | 515 |

OPERATING 8760 HOURS PER YEAR.

| Price of Coal per Ton. | Cost of Brake Horsepower, Year. | Critical Temperature Difference, Degrees Fahrenheit. | Minimum economical temperature of gases. Boiler pressure = 150 pounds gage. Degrees Fahrenheit. |
|---|---|---|---|
| $2.00 | $34.10 | 133 | 499 |
| 2.50 | 41.70 | 109 | 475 |
| 3.00 | 49.40 | 92 | 458 |
| 3.50 | 57.10 | 80 | 446 |
| 4.00 | 64.70 | 70 | 436 |
| 4.50 | 72.40 | 63 | 429 |
| 5.00 | 80.00 | 57 | 423 |

The same theory can be applied to an economizer fed with cold water. Assume that the economizer, installed, costs $1 per square foot, and that the annual charges for interest, depreciation, maintenance, power and attendance amount to 12 per cent.; from which the lowest economical temperature difference with coal costing $3 per ton and the plant running at full load 24 hours per day and 365 days per year, is found to be 32.4 degrees Fahrenheit. With the cold end of the economizer receiving water at 60 degrees Fahrenheit the final flue-gas temperature could be brought as low as 92.4 degrees Fahrenheit. In Table 3 the values of $D$, the economical temperature difference, for the economizer, are given for different prices of coal ranging from $2 to $5 per ton, it being assumed that the coal is high grade and contains 14,500 B.t.u. per pound.

TABLE 3. FINAL TEMPERATURE OF FLUE GASES FOR DIFFERENT PRICES OF COAL FOR AN ECONOMIZER PLANT OPERATING 10 HOURS PER DAY AND 310 DAYS PER YEAR.

| Price of Coal. | Cost of Brake Horsepower, Year. | Critical Temperature $D$, Degrees Fah. | Temperature Flue Gases Leaving Economizer for Various Temperatures of Feed Water, Degrees. | | | |
|---|---|---|---|---|---|---|
| | | | 60 Deg. | 100 Deg. | 150 Deg. | 210 Deg. |
| $2.00 | $14.25 | 112.2 | 172.2 | 212.2 | 262.2 | 322.2 |
| 2.50 | 16.98 | 94.5 | 154.5 | 194.5 | 244.5 | 304.5 |
| 3.00 | 19.68 | 81.4 | 141.4 | 181.4 | 231.4 | 291.4 |
| 3.50 | 22.40 | 71.5 | 131.5 | 171.5 | 221.5 | 281.5 |
| 4.00 | 25.10 | 63.8 | 123.8 | 163.8 | 213.8 | 273.8 |
| 4.50 | 27.80 | 57.5 | 117.5 | 157.5 | 207.5 | 267.5 |
| 5.00 | 30.55 | 52.5 | 112.5 | 152.5 | 202.5 | 262.5 |

FOR A PLANT OPERATING 24 HOURS PER DAY AND 365 DAYS PER YEAR.

| | | | | | | |
|---|---|---|---|---|---|---|
| $2.00 | $34.10 | 47.0 | 107.0 | 147.0 | 197.0 | 257.0 |
| 2.50 | 41.75 | 38.3 | 98.3 | 138.3 | 188.3 | 248.3 |
| 3.00 | 49.40 | 32.4 | 92.4 | 132.4 | 182.4 | 242.4 |
| 3.50 | 57.10 | 28.0 | 88.0 | 128.0 | 178.0 | 238.0 |
| 4.00 | 64.70 | 24.7 | 84.7 | 124.7 | 174.7 | 234.7 |
| 4.50 | 72.40 | 22.10 | 82.1 | 122.1 | 172.1 | 232.1 |
| 5.00 | 80.00 | 20.0 | 80.0 | 120.0 | 170.0 | 230.0 |

The value of the least profitable temperature difference can be expressed algebraically as follows:

$$D = \frac{E}{U \times Y_e \left[\frac{B \times S}{33{,}300 \times Y_b} + \frac{C}{2000 \times N_s \times H}\right]}$$

$B$ = Annual charge in dollars on 1 square foot of boiler surface, including interest, maintenance, depreciation and labor,

$E$ = Annual charge in dollars on 1 square foot of economizer surface, including interest, maintenance, depreciation and labor,

$Y_b$ = Hours per year that the boilers are operated,

$Y_e$ = Hours per year that the economizers are operated,

$C$ = Cost of coal per ton in dollars, including handling of the coal and ashes,

$H$ = Heating value of the coal in B.t.u. per pound,

$S$ = Square feet of boiler-heating surface per boiler horsepower developed,

$N_s$ = Efficiency of the boiler when operated at $S$ square feet per boiler horsepower developed,

$U$ = Coefficient of transmission per square foot per hour per degree difference of temperature between the gases and the water.

In applying these figures practical considerations should not be forgotten; that is, it will not pay to reduce the flue-gas temperature below 250 degrees Fahrenheit, as the resulting condensation and deposits of the water vapor upon the tubes, together with the sulphurous gases, will produce rapid corrosion.

# Operation of the Steam Loop

By F. L. Johnson

*A simple explanation of the principles upon which the action of the steam loop depends, showing how steam may be made to circulate rapidly in a closed system of piping and by this circulation return all condensation to the boiler from which the steam comes.*

If a closed vessel or receiver be placed above a boiler and connected to it by two pipes, as shown in Fig. 1, it is evident that the pressure in the receiver will be the same as that in the boiler and the water level in the pipe leading from the water space in the boiler will coincide with the water level in the boiler.

If the valve on the outlet of the receiver be opened, the pressure in the receiver will be reduced by the outflow of steam and there will be a flow of steam from the boiler to the receiver. The reduction of pressure in the receiver will extend downward decreasingly in the pipes toward the boiler. In one of the pipes the water will rise to a hight depending upon the difference in the pressure acting on the surface of the water

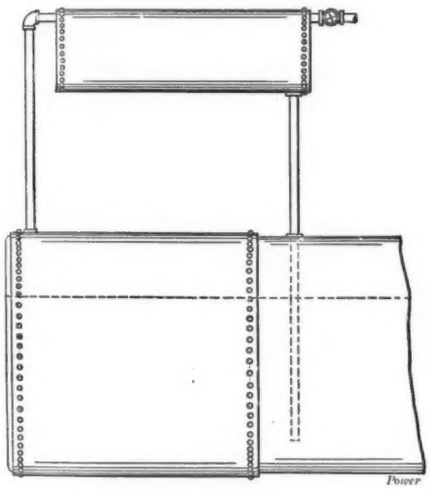


FIG. 1. SHOWING PRINCIPLE OF OPERATION

in the boiler and that acting on the surface of the water in the pipe leading to the receiver.

Through the other pipe steam will rise in an effort to make up the same difference in pressure, but the steam being so much lighter than the water, it will flow to the receiver while the water will only rise in the pipe until it reaches a hight where the weight of that portion above the boiler water level will be equivalent to the difference in pressure betweeen the boiler and the receiver.

So long as the pressure in the receiver is below that in the boiler the flow of steam to the receiver will continue and the water in the other pipe will stand at a hight above the water level in the boiler that will balance the difference in pressures in the boiler and in the receiver.

If the steam which flows to the receiver is wet or holds water in suspension, this water will be swept along in the current of steam until it reaches the receiver, where, owing to the greater volume of the receiver and the consequent slower travel toward the opening, the water will fall to the bottom of the receiver and run into the pipe connected to the water space in the boiler, adding its volume to the column of water already in the pipe, because there is no upward current of steam in this pipe to oppose the downward flow of water. If instead of the pressure in the receiver being reduced by the escape of steam from the outlet it is reduced only by the condensation which takes place, the same flow of steam through one pipe and rise of water in the other will take place, only in a lesser degree because the pressure difference is smaller.

From the action of the steam and water in the boiler and in the receiver, the principles upon which the operation of the steam loop is based may be readily deduced. They are as follows: A difference of pressure may be balanced by a water column; liquids and vapors tend to flow to the region of least pressure; rate of flow is proportional to mass and difference of pressure between two points; decrease of pressure in a steam pipe or vessel is proportional to the condensation; a current of steam will carry water with it by friction.

Like the injector, the steam loop is sometimes called a paradox, while as a

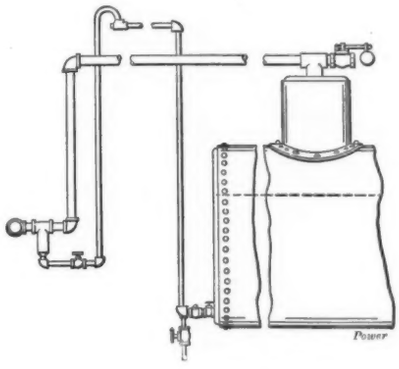


FIG. 2. MAIN PIPE AT BOILER LEVEL

matter of fact its operation and the results produced are just what should be expected from the conditions under which it is installed and operated. It is a system of piping so arranged that water of condensation collected by separators or otherwise is returned to the boiler whence it came without the interposition of traps or pumps. In its ordinary form it is simply a line of pipe consisting of three parts or sections which for convenient reference are called the riser, the horizontal and the drop.

Where the loop is used to return entrainment and condensation from the steam pipe of an engine, a separator is generally placed at or near the throttle valve and the "riser" is connected to the drain pipe of the separator. The riser is carried vertically to a hight depending on the steam pressure carried and the rate of flow desired, and is connected to the "horizontal," which, notwithstanding its name, pitches toward the "drop," into which it discharges. The

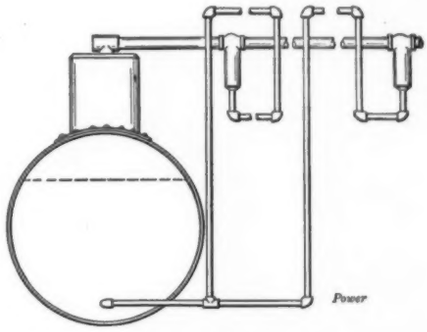


FIG. 3. DOUBLE LOOP ABOVE BOILER

"drop" leads to the boiler below the water line and the water returns through it to the boiler by gravity as fast as its head permits.

If in a system there is a pressure drop of eight pounds from the boiler to the engine, this reduction of pressure will extend through the riser and the horizontal to the "drop," where the difference between the pressure in the horizontal and that in the boiler will be balanced by a water column approximately 20 feet higher than the water line in the boiler.

Now, if condensation in the horizontal reduces the pressure one pound more, the column of water will tend to rise 30 or more inches to balance the difference in pressure between the boiler and the horizontal. But the mixture of steam and spray in the riser will also travel toward this same point and being so much lighter, will travel faster, and as the entrained water reaches the horizontal it runs along to the drop and down to the column rising from the boiler, adding to its hight until it overbalances the difference in pressure, and water flows into the boiler from the bottom end of the drop.

It is the condensation in the horizontal that furnishes the motive power which operates the loop, and when this conden-

sation is maintained at a rate sufficient to give the necessary difference in pressure, the hight of the column in the drop will be that corresponding to the difference in pressure and no greater.

The limits of pressure within which the loop will operate are wide, for the principles of operation are applicable equally to great as well as small differences in pressure and large as well as small quantities of water are readily moved.

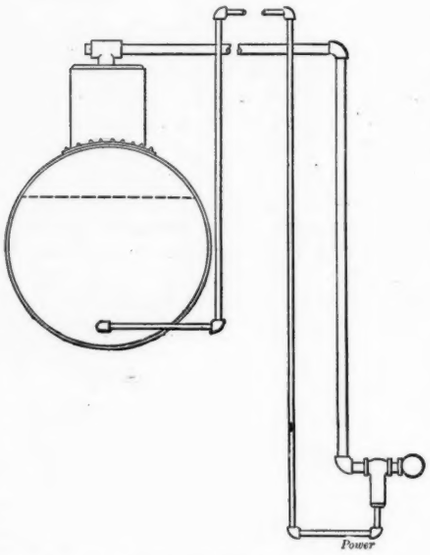


FIG. 4. MAIN BELOW BOILER LEVEL

Water may be returned from a level far below that of the boiler, and the action will continue through long pipes, whether run overhead or laid underground. The use to which the steam is put after the separator has removed the water and the riser conveyed it away has no effect upon the operation of the loop itself, and it may be used to return condensation from all sources, such as heating and drying systems, steam jackets, boiling kettles, etc.

Lack of head room may limit the application of the loop where there is a great difference between the boiler pressure and that at the point of delivery. If the pressure in a return is ten pounds below that in the boiler and a loop 30 feet high is necessary, it would take a loop 150 feet high to return the condensation where there is a difference of 50 pounds.

While the absolute pressure of the steam has nothing whatever to do with the operation of the loop (it will work equally well under low or high pressure), it might fail where the difference in pressure is great as compared with the lowest pressure in the system.

Suppose the boiler pressure is 25 pounds and in the return there is a pressure of only one pound; if the loop is 100 feet high and the condensation in the horizontal is sufficient to produce a vacuum of 14.7 pounds, the water in the drop will rise to about 80 feet, and it is doubtful if the one-pound pressure at the foot of the riser, aided by the vacuum, will force the mixed contents of the riser to the hight of 100 feet. It is certain that it will not if a large percentage of water is present.

Practically, a pressure difference of 15 pounds with ordinary boiler pressure is the condition under which the loop is expected to operate.

As the pressure difference in the system is balanced by the column of water in the drop it makes no difference how far below the boiler the riser starts so long as the pressure at the bottom of the riser is sufficient to lift the mixture of steam and water to the hight of the horizontal.

Fig. 2 is a diagrammatic sketch of the loop as commonly installed for taking care of the condensation and entrainment in a simple installation consisting of a single engine and one or more boilers, with the engine at or near the level of the boiler water line. Fig. 3 represents an installation where the engine is considerably below the boilers.

Two or more separators may be attached to risers which discharge into the same horizontal or separate drops may be connected to the same return pipe, as shown in Figs. 4 and 5.

Fig. 6 illustrates a method of double connection to radiators, drying coils and kettles, by which only dry steam is turned into the system and the condensation goes to the boilers through a second loop.

It may be readily seen from the foregoing description that the steam loop is a simple arrangement of piping which when properly proportioned and connected performs the duties of receiver and return pump. And that once working should continue indefinitely without attention. This it would do if only steam and water were handled, but as all boiler water contains some air which is set free by heat and as not all of it goes through the engine, some will accumulate in the horizontal pipe of the loop and being noncondensible will, if not drawn out of the system, stop its operation.

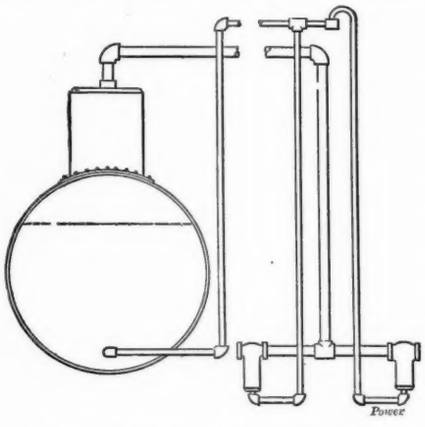


FIG. 5. DOUBLE LOOP BELOW BOILER

Air is a common source of trouble and the probable cause of all failures in operation, so that provision should be made by which the air may be blown out at any time. Sometimes an air cock is placed on the highest point of the piping and sometimes a small pipe is led from the horizontal pipe to a condenser in the plant.

But if the air is removed from the system at frequent intervals by a hand-operated cock or continuously by an open pipe or one leading to a condenser the loop will operate as long as steam for condensation in the horizontal pipe is furnished by the boiler.

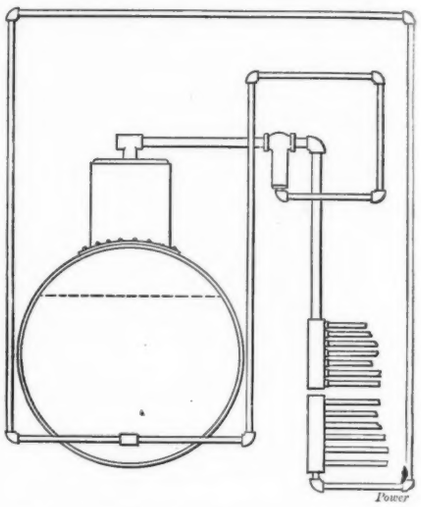


FIG. 6. TWO LOOPS WITH DRY KILN

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It is reported that a second power station of the Shawinigan Water and Power Company will be situated near the present generating plant at Shawinigan falls, on the St. Maurice river, about 85 miles from Montreal, and will be designed for a capacity of 75,000 horsepower, as against 55,000 horsepower in the case of the existing station. It is stated that the company's transmission lines, which are of aluminum carried on wooden poles, are at present the most extensive in Canada, exceeding a total of 400 miles. For transmission to Montreal the current, which is generated at 2200 volts, is transformed up to 50,000 volts; 3000 horsepower is supplied to the Montreal street railways, and 15,000 horsepower to the Montreal Light, Heat and Power Company. It is of interest to mention that the Shawinigan company supply 25,000 horsepower of water power—undeveloped—to the Northern Aluminium Company, and 14,000 horsepower of water power to the Belgo-Canadian Pulp and Paper Company. The greater part of the above power is obtained at a second dam about a mile up the river.

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Do not close the damper entirely when there is fire on the grates, as gas may collect and result in an explosion as destructive, as if caused by steam.

# A Comparison of Efficiencies

By G. A. Glick

*The various types of machines require different methods of computing their efficiencies. The economy of a water wheel and that of a steam engine are figured on a different basis. The author explains how the various efficiencies are obtained.*

One hears sometimes that the efficiency of a waterwheel is 80 per cent. and the efficiency of a steam engine is 10 per cent. The person who hears these per cents. may not have them given to him as a comparison of the poor efficiency of the steam engine and the high efficiency of the waterwheel, but if he is not acquainted with the nature of these efficiencies such might be his impression. Such is not the case. Each efficiency is computed on a different basis.

### Definition of Efficiency

Efficiency is the ratio, usually expressed as a per cent., of the energy utilized to the energy applied in any system, machine or combination of machines. A general expression for efficiency is represented by the equation

$$Efficiency = \frac{E - E'}{E}$$

where $E$ equals energy in the working substance as it enters the machine, and $E'$ equals energy in the working substance as it leaves the machine.

The efficiency as defined above is not for a practical machine, however, and the equation gives the efficiency that is the highest that can be hoped for under the conditions. Some energy is utilized in overcoming losses in the machine itself, and a more practical definition of the term is as follows: Efficiency is the ratio of the energy obtainable from a machine to the actual energy supplied and is represented by the equation

$$Efficiency = \frac{E - L}{E}$$

in which $E$ equals the energy applied to the machine, or the energy in the working substance as it enters the machine, and $L$ equals the term that includes all of the losses.

### Mechanical Efficiency

The term mechanical efficiency is applied to a machine to indicate the loss in that particular machine due to mechanical losses as friction. This efficiency is really a simple one and is expressed by this equation

$$Mechanical\ efficiency = \frac{Input - friction\ loss}{Input}$$

In a steam engine the brake horsepower divided by the indicated horsepower, expressed as a per cent., gives the mechanical efficiency. The difference between the two is taken as the friction loss. This friction loss is assumed to be constant although experiments have shown that it varies with the load. In determining the mechanical efficiency of any other machine some means must be provided to drive the machine idle, and measure the input at the same time. This input is taken as the friction loss and when taken as a constant it is easy to see that the larger the load on the machine the higher will be the mechanical efficiency.

### Efficiency of Heat Engines

Carnot's cycle is one of a purely theoretical nature and is of great use in the study of the theory of heat. It is the cycle of an ideal engine and mathematics will show that the efficiency as calculated by Carnot is the limit toward which the practical heat engines may tend. The efficiency as given by this cycle is expressed by the following equation.

$$Ideal\ or\ Carnot's\ efficiency = \frac{T - T'}{T}$$

in which $T$ equals the upper limit of absolute-temperature range and $T'$ equals the lower limit of absolute-temperature range. Degrees Fahrenheit are changed to degrees absolute by adding 461.

To apply the above efficiency equation assume a perfect heat engine to be operating on Carnot's cycle and receiving steam at a pressure of 200.3 pounds, gage, which it exhausts at atmospheric pressure. The absolute temperature due to 200.3 pounds gage pressure is

$$387.9 + 461 = 848.9.$$

The absolute temperature of the steam at atmospheric pressure is

$$212 + 461 = 673.$$

Substituting these values in the equation, we have

$$Ideal\ efficiency = \frac{848.9 - 673}{848.9} = 20.7$$

per cent. which is the highest that can be hoped for under these conditions.

The high efficiency of the gas engine is due to the high explosion temperature and relatively low exhaust temperature.

This efficiency of Carnot's is known by several names, such as Carnot's Efficiency, Ideal Efficiency and Thermodynamic Efficiency. Carnot's cycle is a reversible one; that is, heat may be changed to work or work may be changed to heat. The heat motors of today are not reversible engines so the application of this efficiency to the practical heat engines merely indicates the upper limit of efficiency which the designer may hope to approach.

### Thermal Efficiency

The efficiency of the heat motor is expressed as a thermal efficiency. This efficiency is the ratio of the useful work done to the work supplied. If the work of the machine and the heat required to do this work are known, the efficiency is

$$Thermal\ efficiency = \frac{Work}{778\ (B.t.u.)}$$

in which *Work* equals the foot-pounds of energy given up by the machine, *B.t.u.* equals the total heat units supplied to the machine, and 778 equals the constant to change B.t.u. to foot-pounds. The above quantities can be calculated from brake tests applied to a machine and the heat contents of the steam per pound. When the total pounds of water used by the engine is known, the total heat used can be calculated.

This efficiency can also be expressed as follows:

$$Thermal\ efficiency = \frac{2545}{\left(Pounds\ steam\ per\ H.P.\text{-}hr. \times \frac{heat\ content}{per\ pound}\right)}$$

Expressed in words this is 2545 B.t.u., the work equivalent of a horsepower-hour, divided by the pounds of steam per horsepower-hour, and also divided by the heat content of one pound of steam equals thermal efficiency.

This efficiency is very low because the heat content of steam is computed from a point known as absolute zero, which is 461 degrees below Fahrenheit zero. A great deal of the heat content of the steam is not available in the engine. It is just as if a waterfall 100 feet high, but 1000 feet above sea level, had its efficiency computed with regard to sea level as the basis for energy content. Because of this, the logical efficiency applied to heat motors is an efficiency ratio described in the following paragraph.

### Efficiency Ratio

The efficiency ratio is the ratio of the B.t.u. available to the B.t.u. utilized. That is, it is the ratio of the heat equivalent of the work done and the heat required to do that work.

$$Efficiency\ ratio = \frac{Heat\ equivalent\ of\ work\ done}{Heat\ utilized}$$

Expressed in other words this efficiency ratio is really the ratio between the thermal efficiency and the ideal efficiency.

### Efficiency of a Refrigerating Machine

It may be surprising to some to hear of an efficiency greater than 100 per cent., but such is the case with a refrigerating machine. This machine works on a reversed cycle and its efficiency may

be expressed as follows. This equation is the same as that of the thermal efficiency of a heat motor except that it is reversed.

$$Thermal\ efficiency = \frac{778\ (B.t.u.\ supplied)}{Work}$$

Now the heat taken up by a refrigerating machine may be several times the heat equivalent of the work done in compressing the working substance, so the efficiency may be several times 100 per cent.

### Efficiency of a Waterwheel

The efficiency of the waterwheel is really an efficiency ratio, but strictly speaking it should not be called such. The efficiency ratio of the steam engine and heat motors gives a fair way of comparing the different motors, but with a waterwheel its location at different hights above sea level makes any efficiency corresponding to the thermal efficiency out of the question. The efficiency below gives a fair means of comparison for all waterwheels. Theoretically 8.8 cubic feet of water per second over one-foot head will develop one horsepower. The expression for waterwheel efficiency is as follows:

$$Efficiency = \frac{P \times 8.8}{q \times head}$$

in which $P$ equals the horsepower developed by the wheel; $q$ equals the water taken by the wheel in cubic feet per second, the *head* equals head, or fall, in feet.

### Efficiency of Dynamo

Electrical efficiency, in general, is also represented by the ratio of output to input. However, the equations for a generator and a motor are as follows:

$$Efficiency\ of\ generator = \frac{Electrical\ output}{Electrical\ output + losses}$$

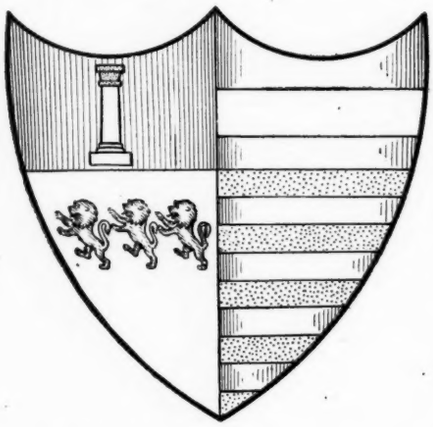
$$Efficiency\ of\ motor = \frac{Mechanical\ output}{Mechanical\ output + losses}$$

or

$$Efficiency\ of\ motor = \frac{Electrical\ input - losses}{Electrical\ input}$$

The losses in electrical machines of these classes are: Copper loss or heating loss in armature and fields; iron losses or hysteresis and eddy currents in armature core, and mechanical losses or bearing friction, brush friction and air resistance to rotating armature.

# The John Cockerill Works at Seraing



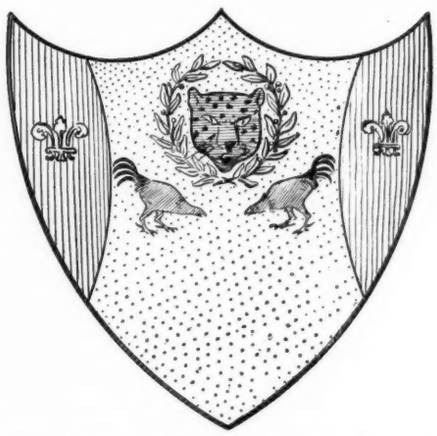
## By F. R. Low

When the use of blast-furnace gas in engines is mentioned one naturally thinks of the great works at Seraing, near Liège, Belgium, which bear the name of John Cockerill. It is generally known that the first large work of this kind was done at Seraing, but how comes it that this large Belgian establishment bears a name so conspicuously English and what has been its history?

John Cockerill was an English emigrant who attracted attention when the mechanical industries which have meant so much for Belgium were in their beginning. He began by manufacturing textile machinery and received in 1817 from the king of Holland for the price of $9000 the château of Seraing and its dependencies with the commission to install there new works and industries. In the works so established the steam engine was applied to the working of iron to an extent before unheard of. The construction of steamboats was undertaken and John Ericsson was here from 1825 to 1827. working out a steamer which must have been provided with high-pressure boilers with internal furnaces, forced draft and with engines having surface condensers, innovations too daring for the epoch. In 1835 the Cockerill works turned out the first locomotive and the first railroad rails used on the continent of Europe. In 1827 they made a centrifugal pump for the Charbonnages des Artistes, at Jemeppe, which machine, entirely forgotten, was still working in 1870. In 1863 they introduced the bessemer process upon the Continent.

The first engine to run upon blast-furnace gas was a small trial machine of eight horsepower, the patents of which the Cockerills purchased from Delamare Debouteville in 1895. Two engines of 250 horsepower each were then built, and are now running in one of the several power stations or "centrals," as they call them, of the works. At the Paris Exposition of 1900, they exhibited an engine capable of developing a thousand horsepower. At the time, its capacity was given as 1000 horsepower with city gas and 700 with blast-furnace gas. Development has shown that the richness of the gas does not make as much difference as was formerly supposed.

Having in mind some recent experiences in the States, I asked the engineer who was conducting me about, how much trouble they had with cylinders breaking. He denied that they had any, and called my attention to their method of cylinder construction. The cylinder and jacket are cast in one piece, no bushing being used, and the metal in the barrel is not over 2 inches in thickness for the largest size, but they are tied lengthwise by enormous bolts passing through the jacket space and tying the back head to the frame in such a way that all the stress is taken by the bolts. They had been free, he said, from piston-rod breakage, which he considered more a question of material than design. The Cockerills have now commenced to build Diesel engines.



The entrance to the works is through the grand court of the old palace, which is maintained as a beautiful garden in the center of which is a fountain, around the base of which are four heroic figures of workmen, said to be portrait sculptures of workmen who emigrated to Belgium with Cockerill.

After the death of Cockerill in 1840 his establishments met with several reverses, but in 1842 they were incorporated with Gustave Pastor, nephew of Cockerill, at the head as the Société John Cockerill, and with a capital of $2,500,000. In 1872 the capital was increased to $3,000,000 and in 1895 cut down to $1,500,000 by the reimbursement to the stockholders of the half of their value. In 1904 it was again raised to $2,500,000. The "Société" now owns its own coal and iron mines and had spent, June 30, 1909, over $20,000,000 for increases and betterments without calling upon its stockholders. The establishments employ between 10,000 and 11,000 hands, the average pay between 1902 and 1909 being about $250 per year.

# The Efficiency of Compressed Air

By Snowden B. Redfield

*"What is the ratio of the work represented in the compressed air to that done by the compressor?" This is a question often asked, and in the present article is answered by the author.*

It is fairly well known that the energy transformation in the use of compressed air is not high but the fact that it "does things," some of them better than by any other means and some which are not done by any other means at all, makes the matter of fuel economy of secondary importance. No apology is needed for the use of compressed air today.

That this matter of the fuel economy is not well understood is evidenced by the repeated appearance of propositions to develop the power of a water fall in the form of compressed air and transmit this through pipes to some distant point where its energy, like that of electricity, is supposed to be turned back again into work.

Such power-transmission propositions as this are utterly absurd because the losses in the compression and expansion of the air reduce the power efficiency to too low a point. These losses are not losses of pressure by friction, nor air by leakage, but simply losses due to heating and cooling.

It is not a difficult matter to prove that all of the work of compression goes into heat and is lost in pipe-line radiation. All of the work performed at the far end comes from the intrinsic heat originally in the air and the economy obtained depends entirely upon how the air is used. The actual cold compressed air traveling in the pipe does not at all represent the energy put into it by the compressor, as this has all been lost in radiation and the only answer to the question first expounded, is: "Nothing."

However, it is known that the air does work and this may be compared to the compressor work (although they have no direct relation) and in this way an "efficiency" may be stated.

As examples of machines using air with little or no expansion, rock drills and pneumatic tools may be cited and some interesting figures as to the efficiency of the power transformation, are given by the accompanying diagrams.

While actually indicating the drill or tool cylinder would be difficult, some rough idea of the indicated horsepower developed within it may be obtained by a little figuring, together with some judicious guessing.

Indicator diagrams of such machines would theoretically be rectangles, but wire drawing and cushioning effects of the valve mechanism would considerably modify this. It may be assumed then, reasoning from such a thing as a steam-pump cylinder, without cutoff, that the diagram factor will be about 80 per cent. In other words the actual mean effective pressure will be about 80 per cent. of what the theoretical rectangular diagram would give.

On this basis it is determined that a standard rock drill having a 3-inch diameter cylinder will develop about 6.2 indicated horsepower with 100 pounds at the throttle, this decreasing with the pressure supplied, down to about 3.7 indicated horsepower, with only 60 pounds pressure.

A 3-inch rock drill will require about 138 cubic feet of free air per minute with 100 pounds pressure at the throttle; this decreasing to 90 cubic feet, with only 60 pounds pressure.

Knowing the quantity of air and the pressure, the compressor horsepower is easily calculated.

Thus, allowing 10 pounds pressure drop in the pipe, a 3-inch rock drill will require 29.8 indicated horsepower in the steam cylinders of the compressor with 100 pounds pressure and single-stage compression, or 25.2 indicated horsepower with compound compression. These figures reduce as the pressure used is reduced, but this, of course, reduces the work done by the tool.

Comparing the probable indicated horsepower developed inside the drill cylinder with the actual compressor power required to furnish the air, gives the probable efficiencies shown by the chart. These efficiencies are referred to both the air and steam cylinders of the compressor, so as to give a basis for calculations for various methods of driving the compressor. They include 10 pounds pressure drop in the pipe line.

Referred to the air end of the compressor, it may be seen that with single-stage compression and 100 pounds pressure, about 23.5 per cent. efficiency is obtained, increasing to about 29 per cent. with the low pressure of 60 pounds. Compound air compression brings these figures up to 27.8 per cent. with 100 pounds and 31 per cent. with 70 pounds.

Referred to the steam end, allowing 88 per cent. mechanical efficiency between the steam and air ends of the compressor, single-stage compression gives a little less than 21 per cent. efficiency with 100 pounds and about 25.5 per cent. with 60 pounds air pressure. Compounding the air cylinders of the compressor increases these figures to about 24.5 per cent. with 100 pounds and almost 27½ per cent. with 70 pounds air pressure.

While these figures for efficiency have been determined for rock drills in particular, they apply equally well to almost any machine using compressed air without expansion. It must, however, be remembered that the figures are based upon indicated horsepower only, both in the drill and the compressor. This is because of the impracticability of measuring the "brake horsepower" of the drill. If, however, brake horsepower efficiency is required, these figures for efficiencies of indicated horsepowers can be multiplied by the mechanical efficiency of the device using the air, say 90 per cent. or 80 per cent., as the case may be.

It is to be noted that the higher efficiencies are obtained with the lower pressures. This is because there is less loss by heating the air during compression and therefore it is advisable to use pressures as low as is consistent with the size and weight of the machine required to do a given amount of work.

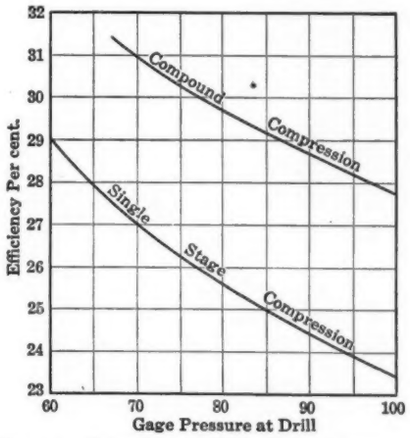



FIG. 1. PROBABLE EFFICIENCY REFERRED TO AIR END OF COMPRESSOR

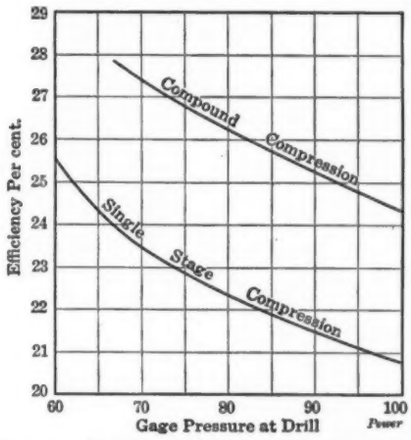



FIG. 2. PROBABLE EFFICIENCY REFERRED TO STEAM END OF COMPRESSOR

# Modern Releasing Valve Gears

By C. A. Tupper*

*The concluding discussion upon the various representative types of releasing valve gear. The earlier developments leading up to present-day practice were discussed in the September 6 issue.*

*Manager of the Reliance Engineering and Equipment Company, Milwaukee, Wis.

The first three gears shown in the following discussion should be credited to George Brown, who proceeded along somewhat different lines from those taken up in the first instalment of this article. This inventor is also responsible for the Brown-Harris gear shown in Fig. 29, which he originated when in the employ of the W. A. Harris Steam Engine Company.

In Fig. 17 is shown the Filer & Stowell gear, for which the principal claims are simplicity, accessibility and noiseless operation. Stress on the valve stem is relieved by the arrangement of the drop lever, the hub of which is so designed as to fit into a recess in the bonnet. The use of double ports in the cylinder reduces the throw of the wristplates and as a whole the gear is well designed for high speeds.

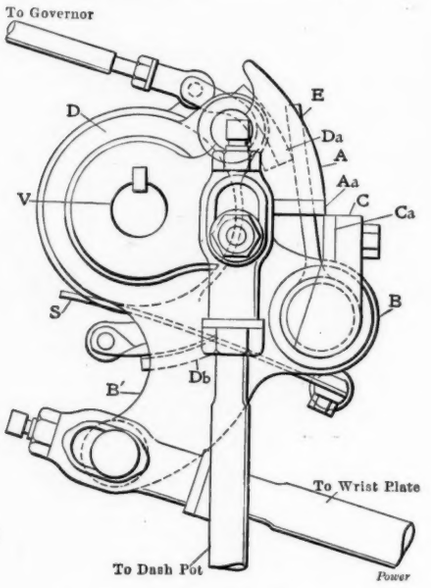



FIG. 17. FILER & STOWELL GEAR

*A* = Drop lever or crank arm, keyed to the valve stem which it actuates, and connected to the dashpot rod.
*Aa* = Catchblock on *A*.
*B B'* = Bell crank which carries the hook *C* on one end of a short shaft journaled in its smaller arm; the other end of the shaft carries a tripping lever *E*. The bell crank receives its motion from the wristplate.
*C* = Hook.
*Ca* = Block carried by *C*.
*D* = Knock-off cam mounted on the valve stem, the position of this cam being determined by the governor.
*Da* = Projection for tripping.
*Db* = Projection for safety stop.
*E* = Trip lever.
*S* = Spring.
*V* = Valve stem.

OPERATION

When *B'* is pulled toward the wristplate, *Ca* engages *Aa*, raising *A* until *E* comes in contact with *Da*. This forces *E* and *C* away, and releases *A*, which is immediately drawn downward by the dashpot and closes the valve.

Fig. 18 is a sketch of the Wisconsin Engine Company's gear, following lines almost identical to those of Fig. 17, with, however, important features of its own. This was fully described in the October 27, 1908, issue of POWER, an abstract of which, for convenience in making comparisons, is given as follows:

In the design of the gear all parts subjected to the greatest strain have been placed as near the cylinder as possible,

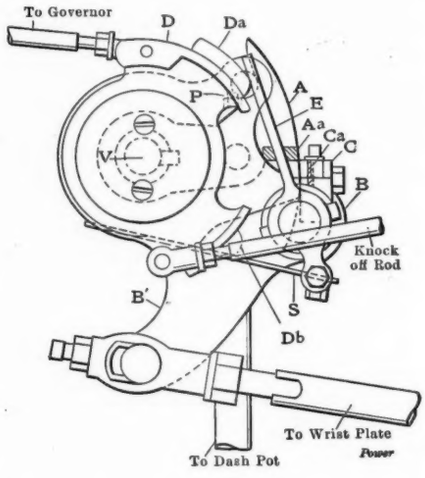



FIG. 18. WISCONSIN ENGINE COMPANY'S GEAR

*A* = Drop lever or crank arm, keyed to the valve stem inside the bonnet opening and connected to the dashpot rod. This has but one-half the ordinary leverage on the bonnet.
*Aa* = Catchblock on *A*.
*B B'* = Bell crank or steam lever, which carries the trip pin *C* on one end of a short shaft journaled in its smaller arm; the other end of the shaft carries a tripping lever *E*. The bell crank receives its motion from the wristplate.
*C* = Trip pin (behind nut and boss).
*Ca* = Block carried by hook.
*D* = Knock-off cam loosely mounted on the bonnet collar, the position of this cam being determined by the governor.
*Da* = Projection for tripping.
*Db* = Projection for safety stop (safety cam).
*E* = Trip lever.
*P* = Closing pin by which the drop lever is forced down should the dashpot fail to work.
*S* = Spring, adjustable while gear is in motion.
*V* = Valve stem supported on both sides of actuating point.

OPERATION

When *B'* is pulled toward the wristplate, *Ca* engages *Aa*, raising *A* until *E* comes in contact with *Da*. This forces *E* away, thus forcing *C* outward and releasing *A*, which is immediately drawn downward by the dashpot and closes the valve.

in order to get the benefit of rigid support and to reduce the leverage of the gear on the bonnet. The drop lever is inside the bonnet, rising and falling through the inside opening in the casting. Besides bringing the valve gear nearer to the cylinder, this construction, which has also been used by Allis-Chalmers Company and other builders in the design of engines for heavy service, offers the advantage of giving a substantial bearing to the valve stem on each side of the point where the turning movement is applied. The steam lever has a long bearing on the bonnet just outside of the drop lever, and carries the trip pin and trip steels by which the drop lever is engaged and the steam valve opened. An annoyance sometimes met with in Corliss engine operation is the wearing away or sudden breaking off of the edges of these steels and the consequent refusal of the drop lever to pick up and open the steam valve; this is obviated in the Wisconsin Engine Company's gear by an arrangement for controlling the lap of these steels while the engine is in motion. The trip pin contains a V-shaped recess, one side of which is held against the adjusting screw by a flat steel spring. By changing the set of this screw the relative positions

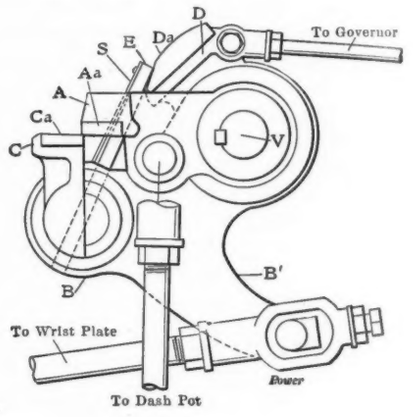



FIG. 19. STANDARD NORDBERG GEAR

*A* = Drop lever or crank arm, keyed on outer end of valve stem which it actuates, and connected to the dashpot rod.
*Aa* = Catchblock on *A*.
*B B'* = Curved double arm loosely mounted on valve stem and receiving its motion from the wristplate; *B* carries hook *C*.
*C* = Hook in form of L-shaped lever carrying die block at upper end.
*Ca* = Block carried by hook.
*D* = Knock-off cam, the position of which is determined by the governor.
*Da* = Projection for tripping.
*E* = Trip lever or arm.
*S* = Spring.
*V* = Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *C* engages *A*, thereby raising *A* until *E* is forced outward by the projection *Da*, carrying with it the hook *C* and releasing *A*, which is immediately pulled down by the dashpot and closes the valve.

of the two steels are varied, so that their laps may be increased when required.

Referring to Fig. 18, the knock-off lever, through which the governor controls the cutoff, is placed at the outer end of the bonnet, where it is practically free from disturbance by the other working parts. There is but one light rod connecting the governor lever with the knock-off lever on the bonnet at the crank end of the cylinder and a light rod connecting the crank-end knock-off lever with that on the head-end bonnet. This not only dispenses with the long rod and its lever between the governor and the head-end bonnet but greatly reduces the weight and consequently the inertia of the parts, which must be moved by the governor to control the cutoff. The governor and gear consequently work in harmony to secure close regulation. In the standard high-speed design no wristplate is used on the steam valves, the motion being transmitted through straight rods direct from the eccentric. Other points are included in the general list of features common to nearly all modern releasing gears, as mentioned near the end of this article.

The gear under discussion is essentially the same as that formerly used by the Brown Corliss Engine Company, which the Wisconsin Engine Company succeeded, and early reference to it will be found under that name.

The Vilter Manufacturing Company, which for many years used the standard hook mechanism, has recently brought out a new gear presenting many features similar to that shown in Fig. 18.

Fig. 19 shows the standard Nordberg gear. Its peculiar feature lies in the form of the hook, which is an L-shaped lever carrying at its upper end a block that engages with the block on the drop lever. The design of this hook, it is claimed, enables a comparatively short crank arm to be used, thus reducing the travel of the hook, and the size of the eccentric to a minimum. The gear, as now built, was described very fully in an article by Osborne Monnett which appeared in the February 22, 1910, issue of POWER.

Fig. 20 illustrates another of the inventions of E. T. Sederholm, now assistant chief engineer of the Nordberg Manufacturing Company. This is a long-range cutoff gear used for many years on the more powerful engines built by Fraser & Chalmers, of Chicago. From the drawing it will be seen that, in one respect, the steam hook is the same as the well known Reynolds type. It is a

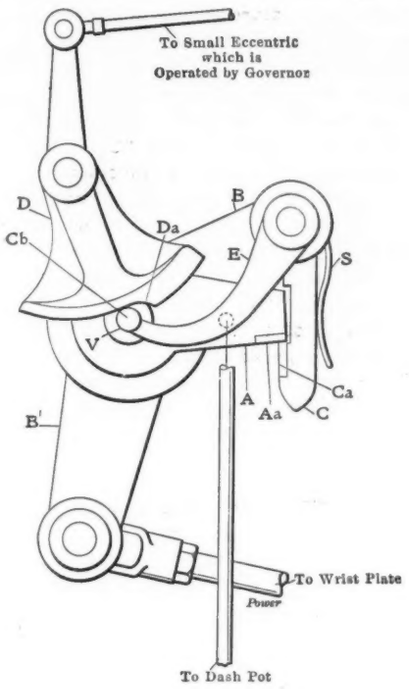



FIG. 20. SEDERHOLM LONG RANGE CUTOFF GEAR

*A* = Drop lever or valve-closing arm, keyed to the valve stem which it actuates, and connected to dashpot rod.
*Aa* = Catchblock on *A*.
*B B'* = Two-arm lever *B* carrying *C* and *E*, and receiving its motion from the wristplate.
*C* = Hook journaled in *B*.
*Ca* = Catchblock on *C*.
*Cb* = Trip roller on lever *E*.
*D* = Knockoff cam, pivoted on the bonnet and positively operated by means of an eccentric.
*Da* = Tripping incline or offset, joining two concentric surfaces on *D*.
*E* = Curved trip arm on *C*.
*S* = Spring.
*V* = Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *Ca* engages *Aa*, raising *A* and opening the valve. Independently of the motion of *B'*, the eccentrically operated knock-off cam *D* releases *A* when *Cb* comes in contact with *Da*. Drop lever *A* is then pulled down by the dashpot, closing the valve. The governor determines the lead of the knock-off cam *D*, thereby changing the admission from no steam to 0.8 stroke.

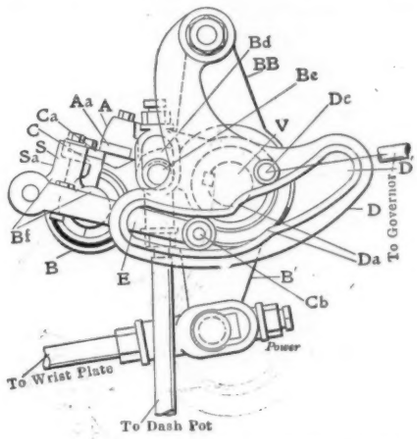



FIG. 21. FULL-STROKE NORDBERG GEAR

*A* = Drop lever or valve-closing arm keyed to the valve stem which it actuates, and connected to the dashpot rod.
*Aa* = Catchblock on *A*.
*B, B', B B* = Three-arm lever formed with a sleeve and turning on the bonnet through which the valve stem passes. *B* carries hook *C*, spring *S* (incased in *Sa*) and arm *Bf* which carries *E* on the stud projection. *B'* is connected to the wristplate. *B B* holds *D* suspended from the stud.
*Bd* = Pivot pin on *A* extending into the path of *Be*.
*Ba* = Projection on the outer sleeve face of triple lever (*B, B', B B*).
*Bf* = Arm connected to stud on *B*, outside of *C* and overhanging *E*, which it holds suspended by the stud on its end and by *S*.
*C* = Hook in form of L-shaped arm, carrying catchblock *Ca* at the upper end and fitted to turn in the sleeve on stud of arm *B*.
*Ca* = Catchblock on *C*.
*Cb* = Trip roller on *E*, which projects into a slot *D'* in cam *D*.
*D* = Knock-off cam in shape of two arcs, each concentric with pivot connection on arm *B B*, and forming their junction, a short incline or offset at *Da* by which the trip arm *E* is turned sufficiently to disengage the latch block *Ca*.
*D'* = Slot in *D* into which *Cb* projects.
*Da* = Offset or tripping point in *D*.
*Dc* = Pivot pin on cam, which connects to cutoff eccentric.
*E* = Trip arm pivoted at one end on the stud which is journaled in *Bf* and carries the roller *Cb* at the other end.
*S* = Spring and bolt (incased) for holding *Cb* engaged to the upper surface of the slot in cam *D*.
*Sa* = Vertical socket formed on the upper side of *Bf* and containing *S*.
*V* = Valve.

OPERATION

When *B'* is pulled toward the wristplate, *Ca* engages *Aa* and carries *A* with it, opening the valve. Independently of the motion of *B'* the eccentric-operated knockoff cam *D* releases *A* where *Da* rides over *Cb*. This occurs exactly when *Dc* passes the center of the valve stem, no matter in what position lever *B B* happens to be. *A* is then pulled down by the dashpot, thus keeping the valve closed. During the return motion *Ca* is kept clear of *Aa* and *A* until ready to engage, thus minimizing noise and wear. The action of the governor is such that it alters the lead of the knock-off cam *D*, thereby changing admission from no steam to 0.8 of the stroke. In case the dashpot should for any reason fail to close the valve, then the return movement of the levers *B, B'* and *B B* causes *Bd* to engage *Be*, thus closing the valve.

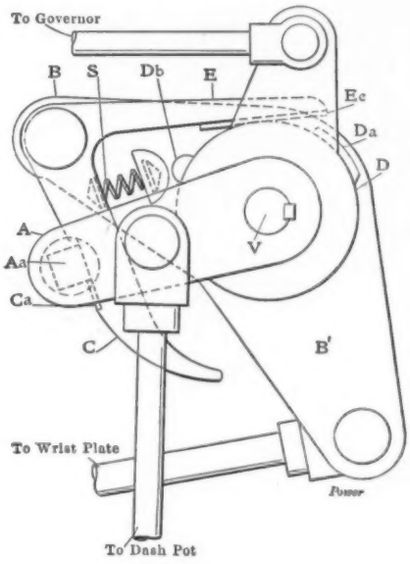



FIG. 22. NEW YORK ENGINE COMPANY'S GEAR

*A* = Drop lever, keyed on outer end of the valve stem which it actuates, and connected to the dashpot rod.
*Aa* = Catchblock on *A*.
*B B'* = Bell crank loosely mounted on outer end of valve-stem bonnet, carrying hook *C*. It receives its motion from the wristplate.
*C* = Hook carried by *B*.
*Ca* = Block carried by *C*.
*D* = Knock-off cam, actuated by the governor.
*Da* = Trip block carried by *D*.
*Db* = Safety block.
*E* = Trip lever.
*Ec* = Block carried by *E*.
*S* = Spring.
*V* = Valve stem.

OPERATION

When *B'* is pulled toward the wristplate, *Ca* engages *Aa* and raises *A* until *Ec* engages *Da*. This forces the trip lever *E* upward and releases *A*, which is quickly drawn down by the dashpot and closes the valve.

drag hook, but instead of having its knock-off portion in one piece with the hook, like the Reynolds gear, it has only the lifting portion of the hook keyed to the inner end of the small rocker shaft mounted on the steam lever. The outer end of this same rocker shaft carries a small curved lever ending with a pin provided with a roller. This pin and roller

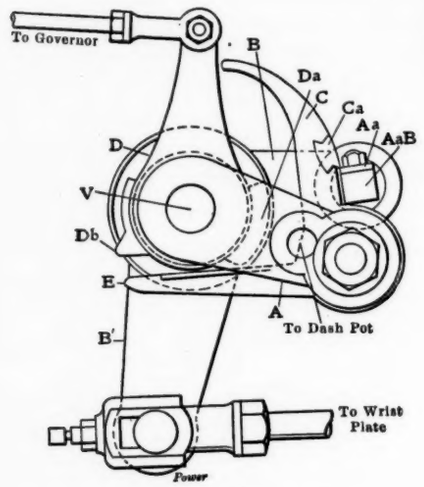



FIG. 23. MINNEAPOLIS GEAR

*A* = Drop lever or steam crank keyed to the valve stem, which it actuates, and connected to the dashpot rod. It also carries claw *C*.
*Aa* = Block on *AaB*.
*AaB* = Catchblock carried by *B*.
*B B* = Bell crank which carries the catch-block *Aa* and rod to wristplate on arm *B'*. This bell crank receives its motion from the wristplate and swings loosely on the valve bracket.
*C* = Claw.
*Ca* = Block on claw.
*D* = Knock-off lever or toe crank which swings loosely on the valve bracket.
*Da* = Steel toe fastened to toe crank (projection for tripping).
*Db* = Safety toe fastened to crank (projection for safety stop).
*E* = Part of claw *C*.
*V* = Valve stem.

OPERATION

When crank *B'* is pulled in the direction of the wristplate, *Aa* engages *Ca*, thereby raising *A* until *E* comes in contact with *Da*. This forces *Ca* away from *Aa* and causes the release of *A*, which is immediately drawn downward by the vacuum pot and closes the valve.

are in the exact center of the bonnet. From this it follows that, no matter in what position the steam lever may be, this pin remains stationary except so far as it turns around on its own axis. The knock-off cam, however, is not stationary but is mounted on a projection of the bonnet and is operated by means of a small separate eccentric. In this manner the cutoff can take place whether the steam valve is opening or closing; in other words, in whatever position of valve gear and piston that may be desired. A cutoff up to 0.8 of the stroke can readily be obtained. The variation in the point of cutoff is obtained by means of changing the lead of the knock-off cam.

The Nordberg Manufacturing Company's full-stroke gear shown in Fig. 21 is somewhat similar to the gear just described except in two important features. The pin on the knock-off lever on the steam hook is not located in the exact center of the bonnet, but has a slight motion around that center. This motion, however, is counteracted by the fact that the knock-off lever is also mounted on a steam lever and, therefore, has exactly the same angular motion as the pin just mentioned. From this it will be evident that in this gear it is immaterial whether the valve is opening or closing. The other important difference is that instead of the knock-off cam being

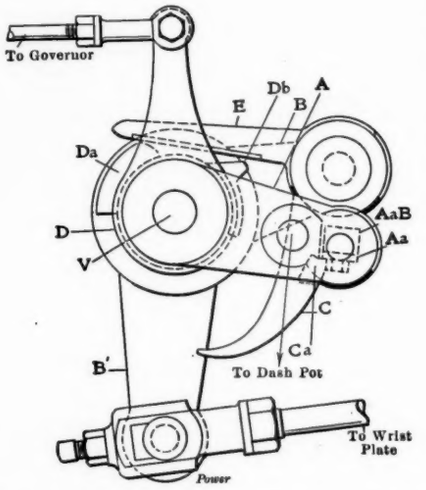



FIG. 24. MINNEAPOLIS GEAR INVERTED FOR HIGH SPEEDS

*A* = Drop lever or steam crank keyed to valve stem which it actuates, and connected to the dashpot rod.
*Aa* = Block on *AaB*.
*AaB* = Catchblock on drop lever *A*.
*B B'* = Bell crank which carries claw *C* and a rod to the wristplate. This bell crank receives its motion from the wristplate and swings loosely on the valve bracket.
*C* = Claw (hook).
*Ca* = Block on claw.
*D* = Knock-off lever or toe crank which swings loosely on the valve bracket.
*Da* = Steel toe fastened to the toe crank (operating cam).
*Db* = Safety toe fastened to the toe crank (safety cam).
*E* = Part of claw *C*.
*V* = Valve stem.

OPERATION

When crank *B'* is pulled in the direction of the wristplate, *Ca* engages *Aa*, thereby raising *A* until *E* comes in contact with *Da* and forces *Ca* away from *Aa*, causing the release of *A*, which is immediately drawn downward by the dashpot and closes the valve.

merely a one-sided cam, it is an inclosed cam, positively holding the pin of the knock-off toe. By this means the knock-off cam not only disengages the gear but it also engages the hook again at the right time, the idea being to do away with the rubbing friction and wear found in all types of Corliss gears where the hook is pushed into engagement by means of a spring.

Fig. 22 shows a gear designed by the New York Engine Company, but as that concern builds few Corliss engines, this has been very little used.

In Fig. 23 is illustrated the gear used by the Minneapolis Steel and Machinery Company on engines running at moderate speeds, and Fig. 24 shows the same gear inverted for high-speed Corliss engines. This is an adaptation of the Corliss crab claw.

Fig. 25 is a sketch of the Atlas gear, the mechanism of which is so simple as to require little or no explanation. The principal feature of interest consists in the form of the hook.

Figs. 26 and 27 illustrate the Corliss gears designed by the Southwark Foundry and Machine Company, which are designated as "Style A" and "Style B," respectively. These are modifications of the standard hook type, the difference between them being plainly shown by the sketches. The closing action of "Style B," being positive, renders it particularly suitable for high speeds.

In a class by themselves are the gears shown in Figs. 28 and 29. The first of these is the gear originally used by the Wm. A. Harris Steam Engine Company and the second is the mechanism which the writer

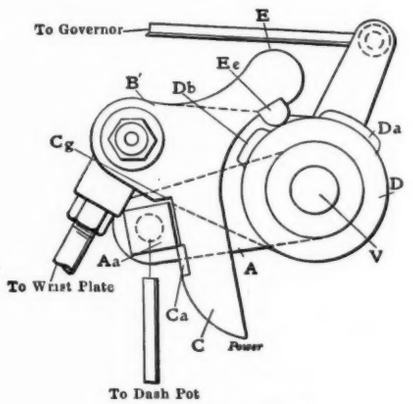



FIG. 25. ATLAS GEAR

*A* = Drop lever or crank arm mounted on the valve stem, which it actuates, and connected to the dashpot rod.
*Aa* = Catchblock on *A*.
*B'* = Arm shown by dotted lines, which takes the place of the usual bell crank. This arm receives its motion from the wristplate.
*C* = Hook carried by *B'*.
*Ca* = Block carried by hook.
*Cg* = Safety stop.
*D* = Knock-off cam, the position of which is determined by the governor.
*Da* = Projection for tripping.
*Db* = Projection for safety stop.
*E* = Trip lever.
*Ee* = Projection on hook for tripping.
*V* = Valve stem.

OPERATION

When *B'* is pushed away from the wristplate, *Ca* engages *Aa* and raises *A* until *Ee* comes in contact with *Da*, thus forcing *E* outward and causing the release of *A*, which is immediately drawn downward by the dashpot and closes the valve. Should for any reason the valve fail to close, the return movement of the lever *B'* will cause *Cg* to close the valve.

understands to be now employed. It is claimed for the gear illustrated in Fig. 29 that it is one of the simplest releasing mechanisms used on Corliss engines at the present time, and that the absence of springs, links, etc., tends to make it very smooth running. The movement of the hook is positive; and, as the center upon which the releasing bar turns is below the plane of contact of the blocks, the disengagement is so easy as to cause no perceptible jar on the lever.

This device, or one very similar to it, has also been used by the Harrisburg Foundry and Machine Works; but, in place of the ordinary dashpot used to close the steam valve, the gear, as designed by this company, was connected with a dashpot having a steam cylinder attached. Thus the closure was effected

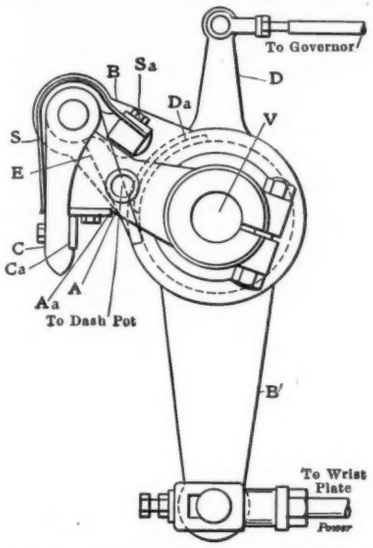



FIG. 26. SOUTHWARK GEAR, STYLE "A"

*A* = Drop lever or crank arm mounted on the valve stem and connected to the dashpot rod.
*Aa* = Catchblock on *A*.
*B B'* = Bell crank which receives its motion from the wristplate or directly from the eccentric.
*C* = Hook or pawl carried at extremity of *B* on a short shaft journaled in *B B'*. *C* is normally held by the spring *S* against a projecting face on *A*, radial to the center of the stem.
*Ca* = Block on hook.
*D* = Knock-off cam mounted on a bracket. The position of this cam is determined by the governor.
*Da* = Projection on *D* for tripping.
*E* = Trip lever mounted on the same shaft with hook *C*, on the opposite side of *B*.
*S* = Spring.
*Sa* = Spring block.
*V* = Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *Ca* engages with *Aa* and causes *A* and *B B'* to move in unison until *E* comes in contact with *Da* on *D*; *E* is then pushed outward and with it *C*, thereby releasing *A*, which is immediately drawn downward by the dashpot and closes the valve. When the governor reaches its highest position the projection *Da* holds *E* in its raised position and prevents *Ca* from engaging *Aa* and keeps the valve closed. When the governor is in the lowest position, *Da* does not touch *E* and release *A*. The spring block *Sa* closes the valve if the dashpot fails to do so.

by steam, instead of gravity or vacuum, enabling the engine to be run at higher speeds than were then considered permissible with the ordinary Corliss gear. The use of this mechanism has, however, been discontinued since the introduction of the so called four-valve engine perfected by B. T. Allen, chief engineer of the Harrisburg company.

Fig. 30 is a reproduction of the well known Bass gear, which, like the two gears just described, is of a type peculiar to itself. The construction and details of operation are clearly shown by the diagram and accompanying explanation. This gear is designed to be dust proof. The valve stems are relieved from any stress, which is borne by the bonnets, practically the same object being accomplished as in the standard gears previously described. The working parts of the gear are very accessible for adjustment and removal, and the absence of the usual hook permits the use of larger wearing surfaces, such as the trip blocks present, than could otherwise be obtained in the same space.

Fig. 31 represents the first of the cam and roller gears to be treated in this article. It was designed for the Philadelphia Engineering Company's engine. The action of the mechanism may be plainly understood from the drawing and accompanying illustration. It combines simplicity with noiseless operation.

A mechanism of somewhat similar type

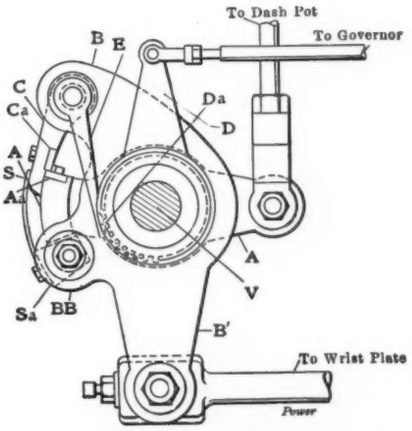



FIG. 27. SOUTHWARK GEAR, STYLE "B"

*A* = Drop lever or crank arm mounted on the valve stem and connected to the dashpot rod.
*Aa* = Catchblock on *A*.
*B, B', B B*= Three-arm lever which receives its motion from the wristplate, or directly from the eccentric.
*C* = Hook or pawl carried at extremity of *B* on a short shaft journaled in *B*, the opposite end of which carries the trip lever *E*. *C* is normally held against *A* by the spring *S*.
*Ca* = Block on hook.
*D* = Knock-off cam lever mounted on bearing supporting the gear. The position of this cam lever is determined by the governor.
*Da* = Projection for tripping.
*E* = Trip lever mounted rigidly on the same shaft with *C*.
*S* = Spring.
*Sa* = Closing block on *B B*.
*V* = Valve stem.

OPERATION

When lever *B* is pulled in the direction of the wristplate, *Ca* engages with *Aa* and causes *A* and *B* to move in unison until *E* comes in contact with *Da* on *D*; *E* is then pushed outward and with it *C*, thereby releasing *A*, which is immediately pushed down by the dashpot and closes the valve. When the governor reaches its highest position the projection *Da* holds *E* in its raised position, preventing *Ca* from engaging with *Aa* and keeping the valve closed. When the governor is in the lowest position *Da* does not touch *E* and release *A*. The spring block *Sa* closes the valve if the dashpot fails to do so.

is the Fishkill gear, illustrated in Fig. 32. With the action of the rollers there is no appreciable strain in turning the triple lever on its axis and consequently no tendency to disturb the normal action of the governor.

Fig. 33 illustrates the details of the Watts-Campbell gear, the principal merit of which is its simplicity and noiseless operation.

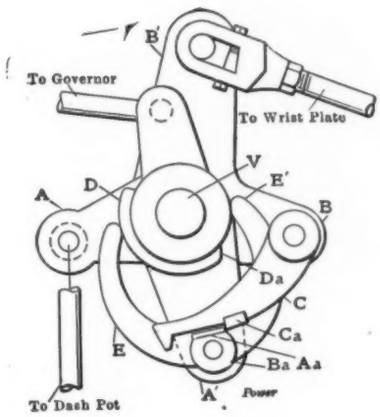



FIG. 28. ORIGINAL HARRIS GEAR

*A A'* = Drop lever or bell crank keyed to the valve stem which it actuates, and connected to the dashpot rod.
*Aa* = Catchblock carried by *Ba*.
*Ba* = Sleeve carried by a pin at the end of *A'* and holding a catchblock on top.
*B B'* = Bell crank receiving motion from the wristplate.
*C* = Pawl which takes the place of the usual hook.
*Ca* = Block carried by *C*.
*D* = Knock-off cam, loosely mounted on the bonnet, the position of the cam being determined by the governor.
*E E* = Curved arms extending from *Ba* and acting as trip levers.
*V* = Valve stem.

OPERATION

When *B'* is pulled toward the wristplate, *Ca* engages *Aa* and raises *A* until *E'* comes in contact with *Da*. This pushes *E'* outward and turns *Aa* around sufficiently to release *A*, which is immediately drawn downward by the dashpot, thus closing the valve. Should the governor reach its lowest position, *D* would hold *E E* in such a position that *Ca* cannot engage *Aa* and the valve would remain closed. The hook at the end of *C* is also a device to insure the closing of the valve should the dashpot fail to act.

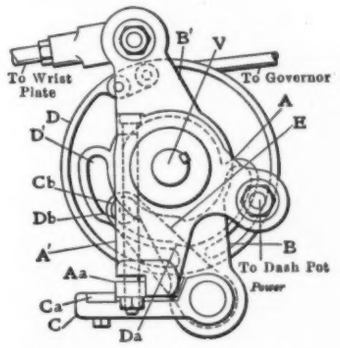



FIG. 29. LATER HARRIS GEAR

*A A'* = Bell crank keyed to the valve stem which it actuates. *A* is connected to the dashpot rod.
*Aa* = Block carried at the end of *A'*.
*B B'* = Two-arm lever loosely mounted on the bonnet and receiving its motion from the wristplate.
*C* = L-shaped hook carried by *B*, and moved by roller running in cam groove *D'*.
*Ca* = Block carried by *C*.
*Cb* = Trip roller on upper extremity of *E*.
*D* = Knock-off cam loosely mounted on the bonnet, the position of which is determined by the governor.
*D'* = Cam-shaped groove.
*Da* = Concentric deviation from the valve stem for tripping.
*Db* = Concentric deviation from the valve stem for safety stop.
*E* = Trip lever, keyed to *C*.
*V* = Valve stem.

OPERATION

When *B'* is pulled toward the wristplate, *Ca* engages *Aa*, raising *A* until *Cb* comes in contact with *Da* and strikes into the lower deviation of *D'*. This forces *C* outward and releases *A A'*, which immediately draws *A* downward by means of the dashpot and closes the valve. Should the governor reach its lowest position, *Cb* would rest in the upper deviation of *D'*, preventing *Ca* from engaging *Aa* and keeping the valve closed.

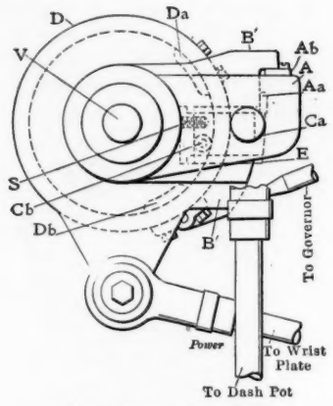



FIG. 30. BASS GEAR

*A* = Drop lever keyed to the valve stem which it actuates, and connected to the dashpot rod.
*Aa* = Catchblock on *A*.
*Ab* = Leather strip on a lateral projection of *A*, to avoid hammering of the bell crank *B'* on the drop lever *A*.
*B B'* = Bell crank receiving its motion from the wristplate.
*Ca* = Block carried by *E*.
*Cb* = Trip roller on *E*.
*D* = Knock-off cam, the position of which is determined by the governor.
*Da* = Projection for tripping on the inner rim of the knock-off cam.
*Db* = Projection for safety stop.
*E* = Sliding trip block, fitted in a recess of *B B'*, which takes the place of the trip lever. This has at its outer end a die block *Ca* and on its inner side a roller *Cb*.
*S* = Spring.
*V* = Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *Ca* engages *Aa* and raises *A* until *Cb* comes in contact with and rides over *Da*; the latter causes *Cb*, and with it *E*, to slide inward toward the valve stem, thereby disengaging *Ca* from *Aa* and releasing *A*, which is immediately drawn downward by the dashpot, closing the valve.

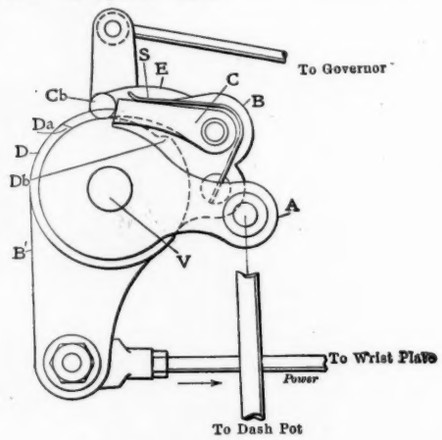



FIG. 31. PHILADELPHIA GEAR

*A* = Drop lever or crank arm, mounted on the valve stem, and connected to the dashpot rod.
*B B'* = Bell crank, which receives its motion from the wristplate.
*C* = Hook or pawl carried at extremity of *B'* on a short shaft journaled in *B*, the opposite end of which carries *E*. *C* is normally held by the spring *S* against the top of *A*.
*Cb* = Trip roller.
*D* = Knock-off cam mounted on the valve stem. The position of this cam is determined by the governor.
*Da* = Projection for tripping.
*Db* = Projection for safety stop.
*E* = Short curved lever carrying a roller *Cb*.
*S* = Spring.
*V* = Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *C* engages *A* and causes *A* and *B B'* to move in unison until *Cb* comes in contact with *Da* on *D*; *E* is then raised, and with it *C*, thereby releasing *A*, which is immediately drawn downward by the dashpot and closes the valve.

In Fig. 34 is shown another type of long-range valve gear, the working of which is similar to that of the gear illustrated by Fig. 21. A separate eccentric is used for operating the knock-off mechanism, but instead of the slotted knock-off lever and floating lever, a bell crank is fastened to the bonnet, one arm of which carries a small roller moving across the center of the valve stem. One leg of the steam hook extends to the center of the valve stem, and whenever the roller crosses the center of the stem the hook is released. This gear was used in the 'eighties on a quadruple-expansion engine built by Edwin Reynolds for the Warren Manufacturing Company, and on several large rolling-mill engines for the Carnegie Steel Company.

The Providence Engineering Works, successors to the Providence Steam Engine Company, and Rice & Sargent Engine Company claim to have been the first manufacturers in the world to introduce a releasing valve gear of the gravity cutoff type, where the engaging mechanism drops into position by the aid of gravity alone rather than by the aid of any springs. This, they state, permitted the old Greene engine to operate at speeds much higher than those that were then the practice. Contemporaneously with the former company, Rice & Sargent developed practically the same idea with an improved method of application. This gear, as at present perfected, is shown in Fig. 35. No wristplate is used; the valve motions are as short as possible and all parts of the releasing mechanism are made very strong with large wearing surfaces. A very clear description of the details and operation of the gear is given in Collins' book on "Valve Setting" heretofore referred to.

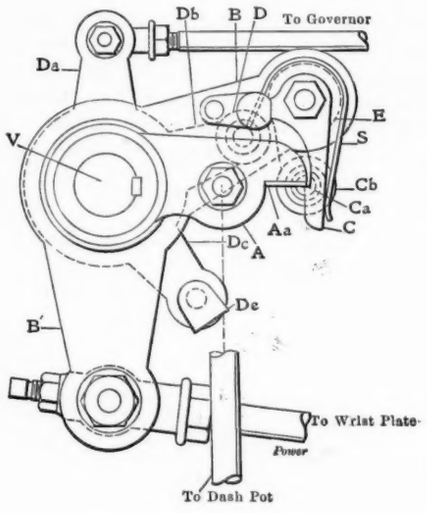



FIG. 32. FISHKILL GEAR

*A* = Drop lever or crank arm mounted on the valve stem which it actuates, and connected to the dashpot rod.
*Aa* = Block carried by *A* and by *C*.
*B B'* = Bell crank carried on a projection of the cylinder bonnet and receiving its motion from the wristplate.
*Da, Db, Dc* = Triple lever oscillating upon a projection of the bonnet, *Da* being connected by an adjustable rod to the governor.
*C* = Hook at the end of a small rock shaft, carried by *B*.
*Ca* = Notch in *C*.
*Cb* = Roller carried by *E*.
*D* = Roller carried by arm *Db* of triple lever at the end of a pin. Its position is determined by the governor.
*Dc* = Adjustable roller carried by arm *Dc* of the triple lever; its position is determined by the governor.
*E* = Lever carried by rock shaft on the side opposite from *C*.
*S* = Spring.
*V* = Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *Ca* engages *Aa*; *A* is carried along an arc concentric with the valve stem, and as *Cb* passes over *D* it is pushed outward, releasing *A*, which is immediately drawn downward by the dashpot and closes the valve. The arm *Dc* of the triple lever carries an adjustable cam *Dc*. Should the governor reach its lowest position *De* would come in the way of the roller *Cb*, which rides on top of it, thus preventing *C* from engaging *Aa*, and keeping the valve closed.

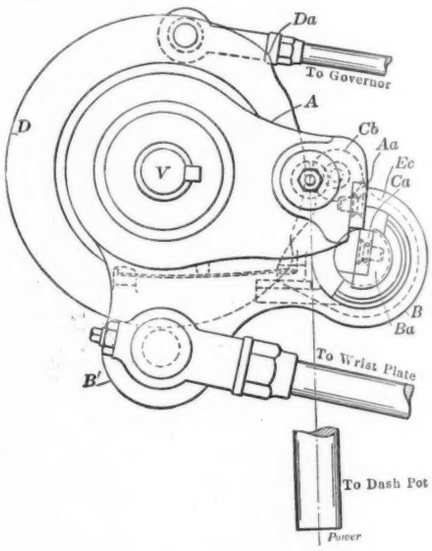



FIG. 33. WATTS-CAMPBELL GEAR

*A* = Drop lever or crank arm keyed to the valve stem which it actuates, and connected to the dashpot rod.
*Aa* = Block carried on the end of *A*.
*B B'* = Bell crank receiving its motion from the wristplate. At the end of *B'* is fixed the pin *Ba*, upon the inner side of which is secured the rocker arm *E C* carrying the roller *Cb*. On the outer side *Ba* carries a die block *Ca*.
*Ca* = Catchblock on *Ba*.
*Cb* = Trip roller on *E C*.
*D* = Knock-off cam, the position of which is determined by the governor.
*Db* = Safety stop.
*Da* = Projection for tripping.
*E C* = Trip lever held against *D* by a spring.
*V* = Valve stem.

OPERATION

When *B'* is pulled toward the wristplate, *Ca* engages *Aa*, raising *A* until *Cb* comes in contact with *Da*, thus forcing *E C* outward and releasing *A*, which is immediately drawn downward by the dashpot and closes the valve.

The Franklin gravity valve gear used by the Hewes & Phillips Iron Works is illustrated in Fig. 36. The latch is a bronze bar hollowed out for lightness and suspended in such a manner that gravity enables it to act properly even at speeds as high as 200 revolutions per minute without the aid of the spring, which is furnished merely as additional security. When the latch blocks engage, the con-

tact of the latch arm takes place upon a fiber block at the side of the upper block, thus avoiding even the clicking sound which usually accompanies the operation of a releasing gear.

Following are features common to the majority of Corliss gears, of which no mention has been made in the preceding descriptions, except where it was desirable to emphasize some special point in order to show the peculiarity of construction.

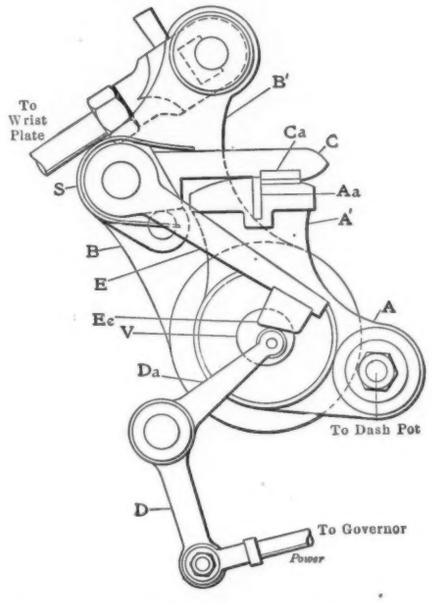



FIG. 34. REYNOLD'S "WARREN ENGINE" GEAR

*A A'*=Drop lever or crank arm mounted on the valve stem which it actuates. *A* is connected to the dashpot rod.
*Aa*=Catchblock carried by *A'*.
*B B'*=L-shaped crank receiving its motion from the wristplate. At the upper end of *B'* the crank is connected to the wristplate; *B B'* also carries hook *C*.
*C*=Hook.
*Ca*=Catchblock on C.
*D Da*=Bell crank connected to the bonnet; lever *D* connects with the governor, while lever *Da*, which carries a small roller at its outer end, acts as the knock-off cam. Its position is determined by the governor.
*E*=Knock-off lever.
*Ec*=Knock-off block carried by lever *E*.
*S*=Spring.
*V*=Valve stem.

OPERATION

When *B'* is pulled in the direction of the wristplate, *Ca* engages *Aa*, causing *A A'* and *B B'* to move in unison until *Ec* comes in contact with roller on *Da*, which pushes *E* and *C* upward. This releases *A A'*, which is immediately drawn downward by the dashpot and closes the valve.

Blocks when used are so offset as to be in contact for only half their thickness, thereby enabling such changes in position as to present eight different edges to wear.

Parts made to standard gage and, therefore, interchangeable.

The use of vacuum dashpots, with dustproof casing and air-regulating valves.

A mechanism designed in each instance with the idea of eliminating reaction on the governor.

Valve connections so arranged that all necessary adjustments can be made while the engine is running.

Wristplate or direct-connected eccentric motion so designed that the steam port may be fully opened very early in the stroke.

Exhaust valves seated by gravity.

Provision for automatically closing steam valves and keeping them shut should the gear get out of order.

Nearly all of the different types of valve gear mentioned in this discussion answer the previous requirements, and so far as actuating the admission valves of the cylinders is concerned, the results are essentially the same. They open the valves in about the same time, and they release in about the same time, and if the vacuum pots or similar arrangements

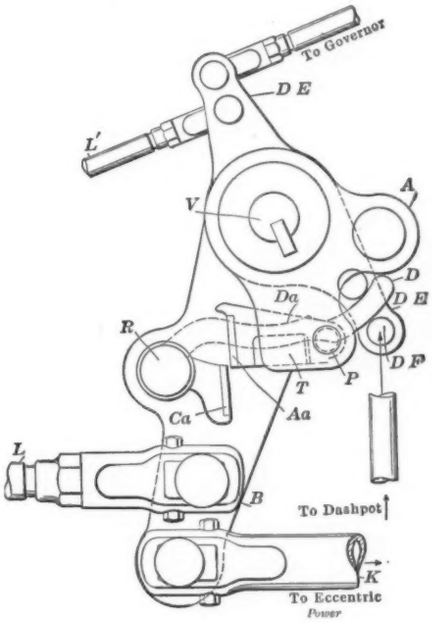



FIG. 35. RICE & SARGENT GEAR

*A*=Drop lever or crank arm keyed to the valve stem which it actuates, and connected to the dashpot rod.
*Aa*=Latchblock resting on and against block *T* in the rear of lever *A*, when at work, but turning on pivot *P* when acting as a latch.
*B*=Crank lever or rocker arm, which carries *Ca*, *D*, *R*, *K* and *L*; it receives its motion through *K* from an eccentric on the engine shaft.
*Ca*=Block on toe lever; the latter is rigidly attached to spindle *R*.
*D*=Cam lever rigidly attached to spindle *R* on rear of arm *B*. This lever is carried between two rolls *D F*.
*Da*=Rise on *D*.
*D E*=Knock-off lever mounted loosely on the valve-stem journal, the upper extremity is connected to the governor rod and the lower holds rollers *D F* on pins. The amount of valve opening and the length of cutoff depend upon the position of this lever, as controlled by the governor—the farther the rollers *D F* are to the left, the earlier the cutoff.
*D F*=Cam rolls which turn on pins in cutoff lever *D E*.
*K*=Connections, etc. (toward eccentric on engine shaft).
*L L'*=Connections to a similar inlet gear at the back end of the cylinder.
*R*=Spindle journaled in arm *B'*. This spindle holds *Ca* and *D*.

OPERATION

When *B* is pulled in the direction of the eccentric, *Ca* engages *Aa*, thereby raising *A* until *Da* passes between rollers *D F*. This forces *D* downward, thus causing *Ca* to release *Aa*: *A* is immediately drawn downward by the dashpot and closes the valve. On the return movement of rocker *B*, cam *D* and toe block *Ca* are raised to the engaging position. Just before the end of this return movement the toe *Ca* raises latch *Aa* sufficiently to let the toe by; the latch turning on its pivot. The latch then drops in front of toe for engagement.

are what they ought to be, the valves will be closed in practically the same time.

The valve gear operated with rollers is apt to make less noise than the one in which the leg of a hook slides over a cam; but, in the writer's judgment, experience has demonstrated beyond question that the valve gear which has the leg of the hook sliding over the cam is more simple and more durable than the other type. The tendency during the last few years, owing to the practice of running Corliss engines at a higher speed, has been toward adopting valve gears with gravity drops and dispensing entirely with springs. Springs, however, insure positive action, whereas the gravity-drop valve gear without springs may not be absolutely reliable under all conditions of work. This is so true that a majority of manufacturers have been slow to adopt it, although present indications are that a number of such gears will be thoroughly tried out in the near future.

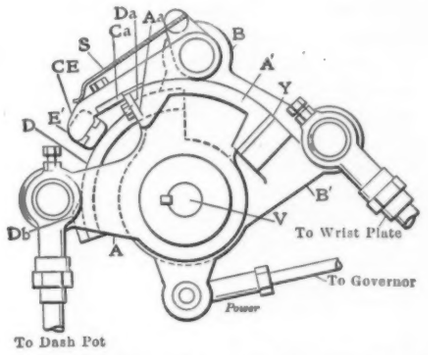



FIG. 36. FRANKLIN GEAR MADE BY HEWES & PHILLIPS IRON WORKS

*A A'*=Drop lever or crank arm keyed to the valve stem which it actuates. *A* is connected to the dashpot rod.
*Aa*=Catchblock on *A'*.
*B B'*=Bell crank which carries a hook and knock-off lever combination *C E*, at the bottom extremity of which the trip block *E* is attached. This bell crank receives its motion from the wristplate.
*C E*=Hook and trip lever.
*Ca*=Catchblock carried by hook.
*D*=Knock-off cam mounted on the valve stem between *A A'* and *B B'*. The position of this cam is determined by the governor.
*Da*=Projection for tripping (operating cam).
*Db*=Projection for safety stop (safety cam).
*E'*=Trip block.
*S*=Spring.
*V*=Valve stem.
*Y*=Safety block on *B'*.

OPERATION

When *B'* is pulled in the direction of the wristplate, *Ca* engages *Aa*, thereby raising *A A'* until *E'* comes in contact with *Da*. This forces *E'* away, causing *C E* to release *A A'*, which is immediately drawn downward by the dashpot and closes the valve. If for any reason the valve should not close, safety block *Y* on *B'* will act.

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On entering the boiler room, ascertain if the valves between the boiler and the water column are open. Then blow down the water column, noting the return of the water in the glass, and test the gage cocks until absolutely sure of the water level. This should be done at the beginning and end of each watch.

# Electrical Department

## Storage Batteries Central Stations and Large Buildings

By Norman G. Meade

**Especially conducted to be of interest and service to the men in charge of the electrical equipment.**

Storage batteries are used in large buildings and central stations for several purposes. In general there is a reduction in coal consumption and operating expenses where they are used, due to the generating machinery being run at the load of greatest efficiency while in service and being shut down entirely during hours of very light load, the battery then carrying the whole of the load. This applies to comparatively small central stations, however, as the large stations operate throughout the entire twenty-four hours. The large companies, however, utilize storage batteries in substations in many instances, which is especially advantageous where the day load consists largely of elevators and similar disturbing machines. By their use it is possible to obtain excellent regulation regardless of the load conditions. In large office buildings also, where many high-speed elevators are in use, storage batteries are used to great advantage in smoothing out the load on the generating equipment.

A smaller generating plant is required where the battery takes the peak of the load, which usually lasts only a few hours but, when no battery is used, necessitates generating equipment of sufficient capacity to give the maximum output, which in some cases is double the normal output. The application of storage batteries to central stations is, of course, confined to direct-current systems or to direct-current distribution mains supplied by rotary converters which take alternating current from a system of high-tension feeders.

A system largely used where the load fluctuates greatly is illustrated diagrammatically in Fig. 1. The armature of a differential booster is connected in series with the battery and the two are connected across the line, as indicated. The shunt field winding is connected across the line and the series field winding in series with the line. At normal load on the generator the current through the series field winding produces a magnetizing effect equal to that of the shunt winding. The magnetism due to the series winding tends to induce in the armature an electromotive force opposing that of the generator, whereas the magnetism due to the shunt winding induces an electromotive force in the same direction as that of the generator.

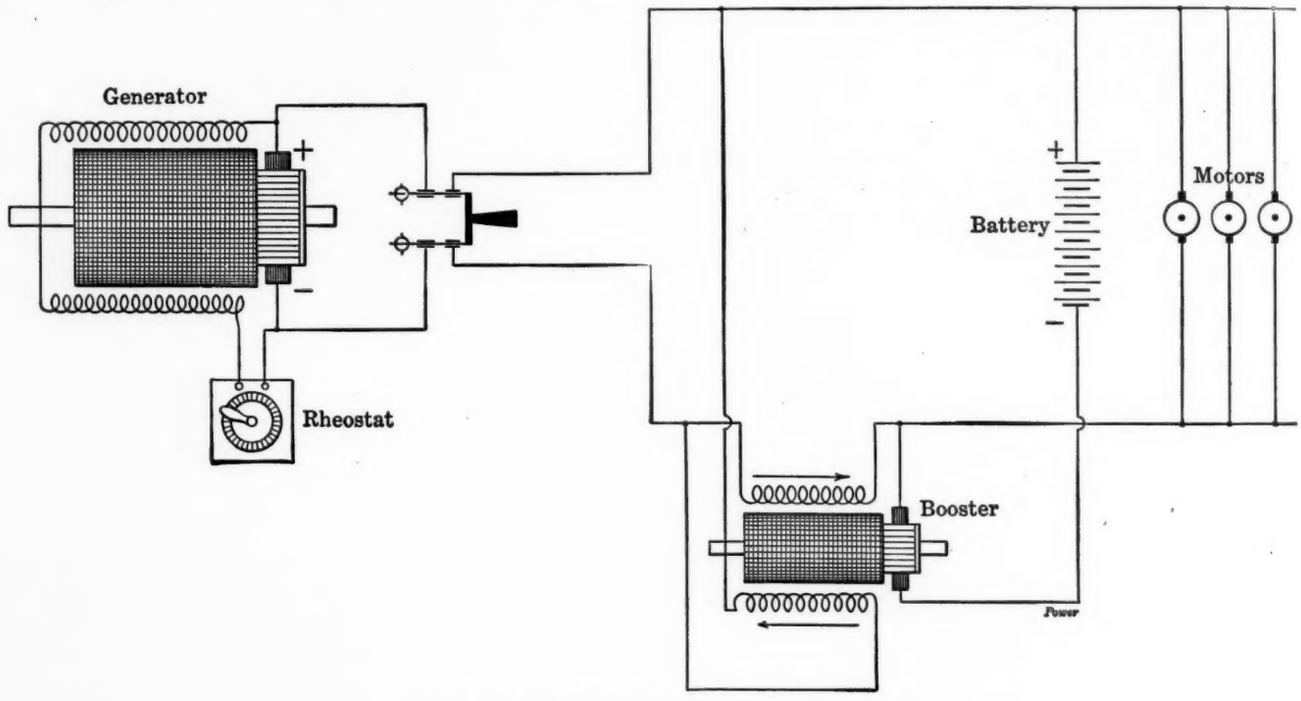



Fig. 1. Automatic Booster System

When the external load equals the nor-

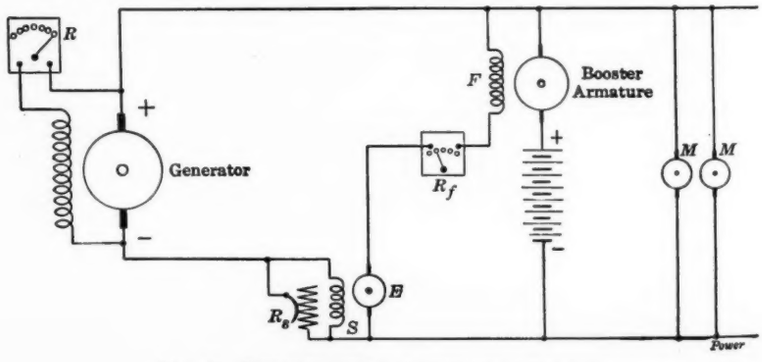



Fig. 2. Booster with Automatic Exciter

mal generator output, no current flows through the battery. An increase above normal load in the main line increases the excitation of the series field winding, which is then stronger than the shunt winding and produces a booster voltage agreeing in direction with that of the battery, which enables the latter to deliver current to the line. If the load should decrease, the diminution in the excitation of the series winding would allow the shunt winding to set up a magnetization such that the booster voltage would assist that of the generator, and current would flow through the battery in the charging direction. By means of this arrangement, therefore, any difference between the normal generator output and the actual load in circuit is absorbed or supplied by the battery as may be required, and the load on the generator is therefore practically constant. There are many varieties of windings for automatic boosters, but they all involve the principle of differential field windings just described.

Automatic boosting systems have been devised in which the field excitation of the booster is varied to meet the requirements of charge and discharge by an external apparatus controlled by the generator current. With these external control devices the booster is a simple shunt-wound generator. Fig. 2 is a schematic diagram of one such system. The external control in this case is effected by means of an exciting generator $E$, the field winding $S$ of which is connected in series with the main generator circuit and shunted by an adjustable resistance $R_s$. The booster field winding $F$ is connected in series with the exciter armature winding and this circuit is connected across the main line. The line potential tends to send current through the field winding $F$ and the exciter armature, and this tendency is opposed by the electromotive force of the exciter armature. When the external load in the main line, represented by motors $M$ $M$, is normal, the excitation due to normal current through the field winding $S$ produces an electromotive force in the exciter armature equal to the line potential; consequently, no current flows in the booster field winding and the booster electromotive force is zero.

With an increase in load there will be an increase in the main line current, and consequently in the field winding $S$; this will increase the exciter electromotive force, which, being then greater than the line electromotive force, will send a current through the field winding $F$ and cause the booster to generate an electromotive force agreeing with that of the battery; this enables the battery to discharge and assists the generator to carry the load. When the load decreases, the excitation of the field winding $S$ decreases and the electromotive force of the exciter falls below the line voltage, which then sends a current through the field winding $F$ and the armature winding of the exciter in a direction opposite to the exciter pressure. This produces a booster electromotive force agreeing with that of the generator, and current passes through the battery in the direction to charge it. Like the differential booster, this system maintains a practically constant load on the generator.

The rheostat $R_f$ controls the booster electromotive force for a given change in the electromotive force of the exciter, and the variable shunt $R_s$ allows a change in the proportion of the generator current through the exciter field winding $S$. In practice, the booster, exciter and motor are all supported on one base and the armatures are mounted on a single shaft.

The regulation of the charging current by means of a shunt-wound booster and the discharge by means of end cells, as described in the previous article* on the application of storage batteries to private plants, is also applicable to central-station service in many cases. Fig. 3 is the complete wiring diagram of such a system applied to three-wire direct-current mains. Two complete outfits, each comprising a booster, a battery and an end-cell switch, are used, one outfit being connected between the positive main and the neutral, and the other between the negative main and the neutral. This combination may be used either to maintain a steady voltage or to equalize the loads in the two divisions of a three-wire system, or both.

*Page 1472, August 16 issue.

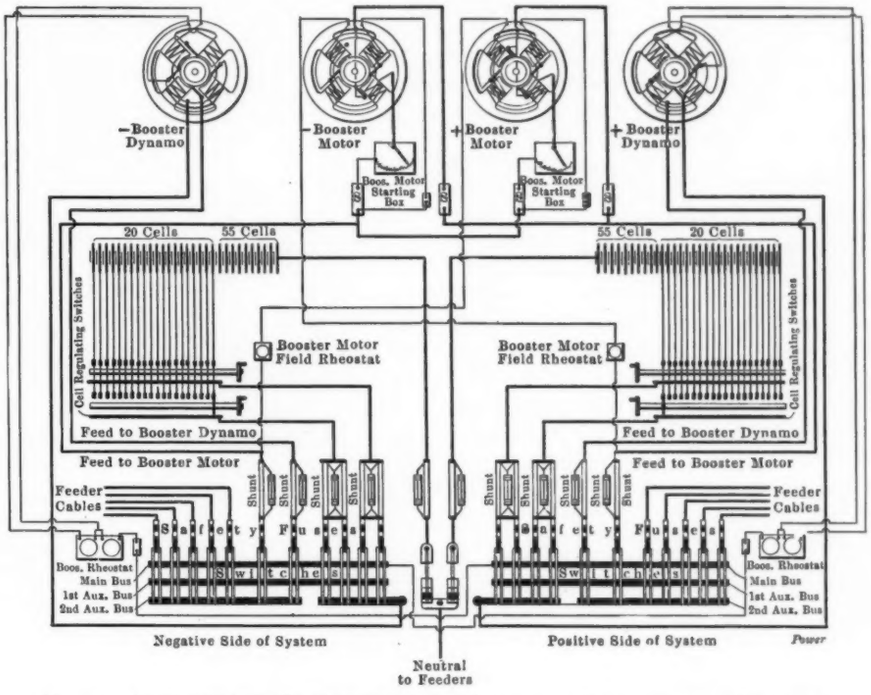
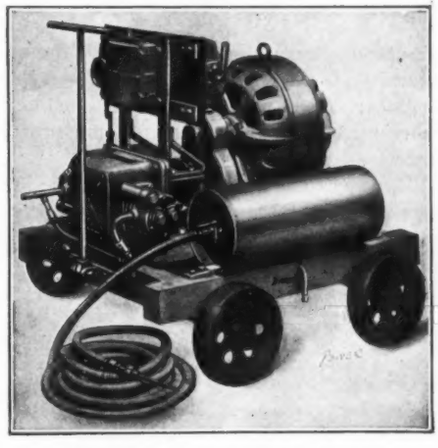



FIG. 3. SWITCHBOARD CONNECTIONS OF COMBINED BOOSTER AND END-CELL REGULATION FOR A THREE-WIRE SYSTEM

## A Portable Motor Driven Air Compressor

The operation of small portable machinery by electric motors has so much to recommend it in the way of convenience and economy that very extended use is now made of the method. The accompanying illustration shows a small portable air-compressor equipment recently supplied to a large flour mill and granary at Rio de Janeiro by the British Westinghouse Electric and Manufacturing Company, Ltd., which, a short time before, carried out the complete electrical equipment of these mills.



PORTABLE COMPRESSOR OUTFIT

This equipment is used for "blowing out" 600-horsepower motors and for cleaning the other machinery around the mills. It consists of a standard Westinghouse air-brake compressor, driven through single-reduction gearing by a 3-

horsepower three-phase squirrel-cage induction motor, mounted, with its control panel, on a substantial wooden truck which is equipped with rubber-tired wheels so that it can be easily moved from point to point. The current for operating it is taken from plug receptacles on the mill walls and conveyed by a flexible cable to a three-pole double-throw inclosed switch on the control panel. The connections are so arranged that when the motor is thrown on the line at starting the fuses are cut out, and after the motor has attained a fair speed, the switch is thrown over to the "running" position, putting the fuses in circuit.

The importance of keeping electrical machinery free from dust cannot be overestimated, and compressed air direct from a compressor is the best form of dust remover, as it blows out all the crevices and cleans the windings thoroughly without impairing the insulation.

## Performance of Railway Motors

In a paper on the Electrification of Suburban Railways, presented at the joint meeting of The Institution of Mechanical Engineers of Great Britain and the

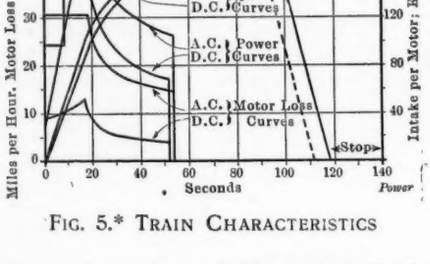



FIG. 5.* TRAIN CHARACTERISTICS

American Society of Mechanical Engineers, F. W. Carter, a prominent English engineer, gave the following interesting data regarding railway-motor performance:

The ultimate temperature rise, in Centigrade degrees, of the hottest accessible part of an ordinary motor, entirely inclosed, after continuous service, is approximately given by the equation

$$T_2 - T_1 = \frac{7\,P}{\sqrt[3]{W^2}}$$

in which $P$ represents the watts lost in the motor (exclusive of the gears) and $W$ is the weight of the machine in pounds, not counting the gears and gear case.

The motor losses in suburban service are about 7 per cent. of the motor intake, with direct-current machines, and 15 to 18 per cent. with single-phase machines.

The heating of the motors has always been a limiting factor in the design of suburban-railway equipment. The use of forced draft for cooling the motors, while practicable for electric locomotives and single cars, is unpractical on multiple-unit trains in suburban service because the equipment cannot be inspected frequently and is therefore too liable to breakdowns.

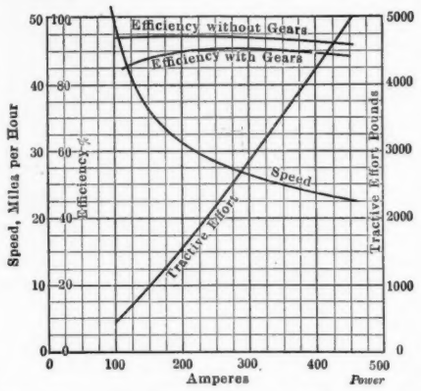



FIG. 6. DIRECT-CURRENT MOTOR CHARACTERISTICS

Fig. 5* gives speed, power and motor-loss curves corresponding to typical suburban schedule runs with direct current and with single-phase equipments, the former of which carries 20 tons per motor and the latter 23 tons. Figs. 6 and 7 show the corresponding motor characteristics. It is evident that the great difference between the systems is in the motor loss, which amounts to 6.75 per cent. of the input in the direct-current case and 17.8 per cent. in the single-phase case. Although both motors are dynamically capable of doing the work, the former alone has sufficient thermal capacity for the purpose, the single-phase motor having to dissipate three times as much energy as it can get rid of by natural cooling.

While it is, perhaps, gratifying to the electrical engineer to reflect that he can propose means fully competent to work the whole railway traffic of this country and can offer three distinct methods of doing it, it is futile, as things are, to regard the problem of electrification from any other point of view than that of the ultimate commercial advantage of the railway companies. The author's investi-

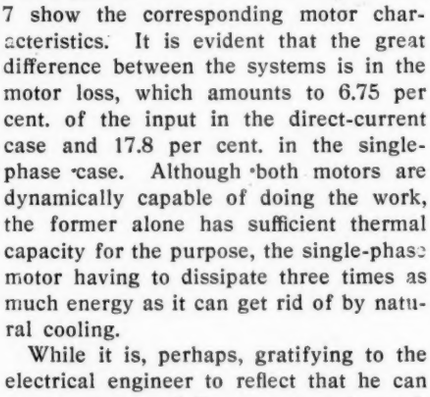



FIG. 7. SINGLE-PHASE MOTOR CHARACTERISTICS

*Figs. 1 to 4 are not shown here.

gations have convinced him that in England, at least, the commercial advantage is only demonstrable in general in the case of heavy suburban service. Even here it should not be taken for granted, for the margin in favor of electrification is by no means overwhelming. This much can be affirmed, however, that where there is any considerable suburban traffic, an investigation of the cost of electrification and the economic results of providing improved facilities will repay the trouble and expense, and in many cases, justify proceeding with the conversion.

The above remarks are intended to apply to the electrification of existing steam-operated lines. The case of entirely new railways is different and much more favorable to electrical operation. In laying out a new branch to an existing railway, again, it may be well worth while to consider electrical operation.

## Economic Results from the Bernon Mills Exhaust Steam Turbine-Generator

In this department last week we published an article by Charles B. Cooke, Jr., describing a method used for electrically locking an exhaust-steam turbine to a reciprocating engine, the exhaust steam of which drives the turbine. Since then we have received the following information regarding the results obtained with the Bernon Mills equipment mentioned in Mr. Cooke's article.

On a weekly run of fifty-eight hours' duration, previous to the installation of the low-pressure turbine, the engine developed an average output of 675 indicated horsepower with a total coal consumption of 116,600 pounds, equivalent to 2.96 pounds per indicated horsepower-hour. The following summary of a test covering an equal period with the low-pressure turbine in operation indicates the striking decrease in fuel consumption that has been effected:

| | |
|---|---|
| Duration of test, hours | 58 |
| Total coal burned, pounds | 98,000 |
| Total water evaporated, pounds | 917,791 |
| Pounds water evaporated per pound of coal | 9.36 |
| Temperature of feed water, Fahrenheit | 102 |
| Average throttle pressure, gage | 82.25 |
| Average vacuum, inches mercury | 28.75 |
| Indicated horsepower of engine: | |
| Right | 224.58 |
| Left | 225.40 |
| Total | 449.98 |
| Turbine power, equivalent indicated horsepower | 486.00 |
| Total power, engine and turbine | 935.98 |
| Coal per indicated horsepower-hour | 1.80 |

If, instead of utilizing the exhaust-steam turbine, this increased output had been obtained with the original economy, the total coal consumed would have been in the neighborhood of 161,000 pounds. It is therefore evident that the saving in fuel alone, due to the low-pressure turbine, closely approaches 31½ tons per week, or about 1650 tons per annum.

# Gas Power Department

## The Gas Turbine

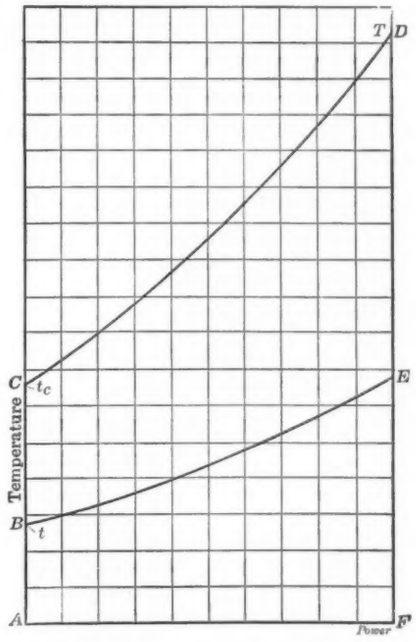
By R. M. Neilson

**Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.**

Prof. Sidney A. Reeve in the issue of Power of July 12 discusses the gas-turbine question and speaks optimistically about its prospects. I have never been pessimistic about the possibilities of the gas turbine, but I have always held and expressed the opinion that an efficient gas turbine cannot be produced until we have an efficient and satisfactory means of compressing air from atmospheric to a high pressure. It is certainly interesting and instructive to compare the different cycles on which a gas turbine could be run; but no matter what cycle is chosen, efficiency cannot, I hold, be obtained without an efficient compressor. There is no sufficiently efficient rotary air compressor at present available, and the employment of a reciprocating compressor of a power great as compared with that of the turbine (and a relatively powerful compressor is essential for efficiency, as will be set forth later on) would take away the greater part of the advantage which a gas turbine might possess over a reciprocating gas engine. The building of an efficient gas turbine is, therefore, in my opinion, an impossibility just now; but there is no obviously insuperable difficulty in the way of producing an efficient rotary compressor. Several persons are working at the subject, and a suitable and sufficiently efficient compressor might be available any day; an efficient gas turbine could then, in my opinion, be built.

I have on several occasions sought to show that an efficient compressor is necessary. As justification for setting forth the reasons again (if justification is necessary), I will try this time to discuss the whole question relating to the factors which limit gas-turbine efficiency more fully than has yet been done, so as to remove any lingering doubt on the subject.

There is a limit to the degree of temperature to which the blades of a turbine can be subjected without rapid erosion, even if cooling arrangements are employed. I have placed the limit at 700 degrees Centigrade (1292 degrees Fahrenheit); Dugald Clerk places the limit considerably lower. It must not be inferred that this limiting temperature is the highest temperature which can be allowed in the combustion chamber or other accessory of the turbine. That is not true, but the blades must not be subjected to a higher temperature. The temperature of the hot products of combustion can be reduced in a divergent nozzle, and the heat energy converted into kinetic energy. Minor difficulties may be encountered in connection with these nozzles; but, in spite of the expressed opinion of Doctor Lucke (whose opinion I nevertheless value), I do not consider these difficulties insuperable. The real difficulty appears when one tries to utilize the kinetic energy in the gas on exit from the nozzle. With a high temperature at admission to the nozzle, the velocity of exit of the gas from the nozzle would be so high that the impact of this gas on a turbine blade would heat the latter above the limiting temperature. It was, I believe, the Hon. C. A. Parsons who first pointed out this difficulty, having previously proved its existence by experiment.




Illustrative Temperature-Entropy Diagram

As an example, it may be said that if air at a pressure of 300 pounds absolute and a temperature of 2000 degrees Centigrade were expanded in an ideal nozzle to atmospheric pressure, the gas would issue from the nozzle at a temperature of 693 degrees Centigrade and a velocity of 5290 feet per second. If gas at this high temperature and enormous velocity were allowed to impinge on a turbine bucket, it would heat the latter much above the limiting temperature, even if the bucket could be given a velocity equal to half that of the gas.

It will not remove the difficulty to compound the turbine for velocity, that is, to absorb the energy of the high-velocity gas jet in several wheels or rings of blades. The first set of blades would probably suffer more in such a case, as there would be a greater difference between its velocity and that of the gas jet.

This temperature objection does not in itself alone prevent the building of a satisfactory gas turbine, but it puts a limit to the temperature of the gas when admitted to the nozzle. If, instead of an inlet temperature of 2000 degrees Centigrade as in the foregoing example, a very much lower inlet temperature, say 1000 degrees C. were adopted, the turbine blades could probably be made to withstand rapid erosion. The exact limit to inlet temperature would require to be carefully investigated, but there is no doubt whatever that if this temperature were sufficiently low, no trouble would be experienced with blades constructed of a suitable material.

*The efficiency of a gas turbine working on any cycle in which the maximum temperature is low is, however, limited by the efficiency of the pump or compressor, and hence an efficient gas turbine cannot at the present day be constructed.*

Proof of the above statement is given in what follows. The whole gas-turbine machine is referred to simply as the "engine"; it comprises a motor which converts the energy in the gas into mechanical work and a pump or compressor which absorbs some of this mechanical work. The meanings of the mathematical symbols are as follows:

$E_i$ = Overall efficiency of ideal engine on any cycle;

$E_a$ = Overall efficiency of actual engine on the same cycle;

$E_r$ = Efficiency ratio $= \frac{E_a}{E_i}$;

$E_m$ = Efficiency of motor alone = ratio of brake work to available energy;

$E_p$ = Efficiency of pump alone = ratio of available energy to brake work;

$W_i$ = Work done by ideal motor;

$W_a$ = Work done by actual motor;

$w_i$ = Work required to drive ideal pump;

$w_a$ = Work required to drive actual pump;

$n$ = Ratio of negative work to gross work in ideal machine $= \frac{wi}{Wi}$.

Then

$$W_a = E_m W_i$$

and

$$w_a = \frac{w_i}{E_p}$$

Therefore,

$$W_a - w_a = E_m W_i - \frac{w_i}{E_p}$$

Now,

$$E_r = \frac{W_a - w_a}{W_i - w_i}$$

and substituting the equivalent of $W_a - w_a$,

$$E_r = \frac{E_m W_i - \frac{w_i}{E_p}}{W_i - w_i}$$

But $w_i = n W_i$ and substituting this equivalent,

$$E_r = \frac{E_m W_i - \frac{n W_i}{E_p}}{W_i - n W_i}$$

which obviously reduces to

$$E_r = \frac{E_m - \frac{n}{E_p}}{1 - n}$$

By definition,

$$E_i = E_i E_r$$

and substituting the last equivalent of $E_r$,

$$E_a = E_i \left\{ \frac{E_m - \frac{n}{E_p}}{1 - n} \right\}$$

It will be evident from the final equation that if $n$ has a high value, $E_a$ is greatly influenced by $E_p$; that is, with a high value of $n$ the overall efficiency of the actual engine is greatly dependent on the pump efficiency, and it is impossible for the engine to have a high overall efficiency if the pump efficiency is low.

For example, suppose that $E_m = 0.8$, $n = 0.4$, and $E_p = 0.5$.
Then

$$E_a = E_i \left\{ \frac{0.8 - \frac{0.4}{0.5}}{1 - 0.4} \right\} = 0$$

That is, no work whatever could be got out of the engine, because the pump would absorb all the mechanical work which the motor gave out; and in this example a high motor efficiency has been assumed, namely 0.8. If the value of $n$ had been put at 0.2 instead of 0.4, the result would have been:

$$E_a = E_i \left\{ \frac{0.8 - \frac{0.2}{0.5}}{1 - 0.2} \right\} = E_i \frac{0.4}{0.8} = \frac{E_i}{2}$$

Such a value for $E_a$ would be quite satisfactory if $E_i$ had a high value. For example, if $E_i$ had a value of 0.7, then $E_a$ would be 0.35, which would be very good; and even lower values of $E_i$ would give values of $E_a$ which would allow a gas turbine to compete with reciprocating gas engines in point of economy.

It is therefore plain that with a non-efficient pump a high overall efficiency could be obtained only if $E_i$ were high and $n$ were low. Now, it is easy to show that with a constant-pressure cycle $E_i$ cannot be high unless $n$ is also high.

The accompanying figure is an entropy-temperature or heat-energy diagram in which the ordinates represent temperature and the areas heat energy; *C*—*D* and *B*—*E* are constant-pressure lines and *B*—*C*—*D*—*E*—*B* represents the cycle of a heat engine working on a constant-pressure cycle; *A* represents the absolute zero of temperature, 273 degrees below the Centigrade zero; $t$ represents the temperature before compression, which will commonly be about atmospheric temperature; $t_c$ represents the temperature after compression and $T$ represents the maximum temperature, all absolute.

With an ideal heat engine working on such a cycle

$$E_i = \frac{t_c - t^*}{t_c}$$

and

$$n = \frac{t_c^*}{T}$$

The value of $t$ is approximately fixed, being about the temperature of the atmosphere, and the value of $T$ is limited by consideration of the erosion of the turbine blades, as previously discussed. To get a high value for $E_i$, $t_c$ must be high; to get a low value for $n$, $t_c$ must be low; it is impossible, therefore, to have a high value for $E_i$ and a low value for $n$ at the same time. Suppose that $t = 290$ and $T = 1300$, absolute temperatures, Centigrade. Then, if $t_c = 870$,

$$E_i = \frac{870 - 290}{870} = 0.6$$

which in itself would be satisfactory; but

$$n = \frac{870}{1300} = 0.67$$

which is impractically high.

If now $t_c$ were made equal to 325, $n$ would be only 0.25, which would allow a gas turbine to do some useful work even with a low pump efficiency; but the value of $E_i$ would be only 0.11, so that $E_a$ would be hopelessly low. It will be obvious on a little consideration that no value of $t_c$ can be found which will be satisfactory as regards both $n$ and $E_i$.

It may be argued, as has been done by Professor Reeve, that a gas turbine need not run on a constant-pressure cycle. That is so, but matters are no better with a constant-volume cycle, in which there is the same impossibility of obtaining a high value for $E_i$ without a high value of $n$; in fact, in no cycle in which the maximum temperature is sufficiently low to be practicable can a high value of $E_i$ be obtained without a high value of $n$. To discuss the different cycles and modifications would involve a repetition of the greater part of the paper which I read before the Institution of Mechanical Engineers (London) in 1904.† Moreover, the reader can prove my statement to be correct by a little scheming with entropy-temperature and pressure-volume diagrams.

*For a proof of these equations the reader is referred to a treatise on thermodynamics.

†"A Scientific Investigation into the Possibilities of Gas Turbines," by R. M. Neilson. *Proceedings*, Institute of Mechanical Engineers (London), October, 1904.

## Desirable Features in a Gas Engine

By L. L. Brewster

No hard and fast rules can be established for the selection of a gas engine, but certain things can be kept in mind and some of them should be indelibly impressed on the memory of any prospective buyer.

While it may not seem so important to the uninitiated, perhaps two of the most important attributes of any gas engine are accessibility and ease of adjustment. This does not mean that gas engines have to be adjusted "every little while" and nursed along to keep them going. It is based on the fact that neglect shows immediate results, which is perhaps an advantage, for it necessitates immediate attention and thereby prevents the "grand smash" which is the cumulative result of persistent neglect. The secret of satisfactory operation in many gas-power plants is the care taken of them, and the easier the job is made, the better it will be done.

For these reasons, it is always advisable to choose an engine in which such parts as the igniter mechanism, valves and pistons can be readily examined, cleaned and renewed, if necessary. A leaky inlet or exhaust valve will cut down economy and decrease power, and where an exhaust valve needs grinding every three months or so it should not be such a heart-rending job as to discourage the engineer into letting it go for six months or a year.

While dirty gas and other unfavorable conditions for which the engine is not accountable are the prime causes of wearing valves, it is found that well cooled and lubricated valves do not show the wear

that uncooled valves do. The latter get overheated, warp out of shape, strike unevenly on the seats and consequently wear more in some places than in others.

The purchaser should remember, when buying a gas engine, that all gas engines are very economical, but their economy in regular operation is right up to the engineer, and for this reason inherent economy is a secondary consideration in the choice of an engine. Convenience in handling and care is much more important.

The lubricating system is a matter of much importance, particularly that serving the cylinders and valves. An inclosed crank case with splash lubrication of the piston cannot be called the best method. The inclosed crank case does not permit the operator to observe the operation as he should and I know of some cases where the inclosed case hid the existence of an overheated main bearing until the resulting smoke escaped through the top of the case; then most of the damage was done.

The proper way to oil the piston is to deliver the minimum practical amount of oil and so time its delivery that it goes in on the piston (preferably between the rings) and is not fed at such times that it will be burned up and do no work. Most of the larger horizontal engines lubricate the piston in this manner and it has proved their salvation in some cases. Whenever possible, particularly in large plants or on large engines, copious lubrication should be supplied to all parts which do not come into contact with the flame, and the lubricating system for these parts should be so arranged as to collect, filter and again use the oil.

The cooling system of any gas engine is highly important. It should be positive in operation and all parts coming into contact with burning gas should be well cooled. It should be made easy for the engineer to observe the quantity and temperature of water flowing to each part and most builders provide for this by making the overflows from all parts visible. Even if well provided with cooling water, the cylinder barrel or head, or both, may be so built as to embody thick masses of metal which are very liable to become overheated, unless an abnormal amount of water is circulated over them. These thick lumps of metal are sometimes the cause of hot spots in the cylinder. Hot spots are one of the causes of "back firing" and back firing sometimes shuts down the engine. Uniform thickness of metal surrounding the cylinder and combustion space is therefore preferable.

When comparing engines of different makes and types do not think that the heaviest engine is always the strongest or best. In the first place, compare actual sizes by comparing piston displacements. Ascertain the total net area of the piston or pistons and this will give an accurate basis on which to compare the power which may be expected from each engine. Some engines are rated at a very high piston speed and may have an actual piston displacement much smaller than others of the same rating, though rated at a lower speed. For this reason piston displacements should be figured for a given piston speed.

Do not be deceived by the given *estimated* weights of engines as set forth in the specifications, for these are not actual but estimates, and the buyer is very liable never to weigh his engine after it is installed. A better and fairer way to compare engines is to compare the relative sizes of crank shafts, crank pins, piston rods, and crosshead pins and shoes if the engine has them, and all these should be stated in percentages of the cylinder diameter, for the reason that unless the engines are all of the same type any other comparison is entirely unfair.

For instance, a single-cylinder, single-acting engine of 100 horsepower, say, should have at least four times the piston area of a four-cylinder single-acting engine of the same power. Consequently the size of the parts in the first engine should be much larger than those of corresponding parts in the latter. Most crank shafts are at least 50 per cent. of the cylinder diameter, and in side-crank engines the best are over 60 per cent. Crank pins are usually made 50 per cent. in the side-crank engine.

It is safe to assume that the character of the service governs very largely the choice of type of an engine. It is quite obvious that for driving electric generators, textile machinery or any other service where uniform speed is essential, the engine should give as nearly uniform a crank effort as possible and should, of course, govern well. Uniform crank effort is usually produced by increasing the number of power impulses per revolution; which means the use of several cylinders or the usual combination of double-acting cylinders in the tandem and twin-tandem forms.

Such arrangements, in combination with good governing, will give results which make possible the operation of engine-type alternators in parallel and this service demands about as uniform speed as any.

If uniform speed is not essential and the power is moderate, it is perhaps better to choose a substantial single-cylinder engine in order to reduce the number of parts. Multiplicity of parts increases the liability to derangement and simplicity is a virtue which in many cases is important, particularly when the class of the attendance is not of the best.

While not a part of the engine, the auxiliaries, which must accompany a gas engine of any size, are important factors in the successful operation of the plant.

The source of ignition current should be reliable and where continuous operation is imperative it should be provided in duplicate.

The air compressor and tank should be of ample size. The tank should have capacity enough to turn the engine over much more than a few times, for it frequently happens in starting that the proper explosive mixture is found only by repeated trials.

# CORRESPONDENCE

## Setting Small Gas Engines

A gas engine requires a very solid foundation. Concrete is universally considered to be the best material for this work, but if the engine is not properly set the firmness of the foundation amounts to very little. I have seen and tried most of the common ways of setting gas engines and have evolved a method which, as far as I know, is original. Although I have used it only for gas engines, it will be found of value in setting other heavy machinery.

In setting an engine, I first build a good solid concrete foundation. After it has had about three or four days to dry, I place the engine on it and line up. Then, by means of as many jacks as may be necessary, I lift the engine as high as possible without letting the anchor bolts get out of the holes in the base, taking care to keep the engine in line with the shaft it is to drive.

Lifting the engine as described gives opportunity to fill the holes around the anchor bolts with cement. After this is done, a thick layer of cement is spread on the foundation where the engine base will rest and the engine lowered into it, leaving the engine level and supported by the jacks until the cement has set. The jacks may then be taken out and the nuts screwed down. If the work is carefully done, the engine and foundation will be as solidly united as if they were in one piece and the engine will run as smoothly as though driven by steam.

In making the anchor bolts I join two rods to form a large inverted U and bed them in the foundation. They cannot pull out or turn around and are cheaper to make than any of the usual forms. The bends at the bottom should not be too sharp, as that would increase the tendency to break at that point.

H. K. WILSON.

New Bedford, Mass.

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When a gas engine stops suddenly, in nearly every instance a loose or broken connection in the ignition system is the cause. Such stops are also caused, however, by the stoppage of the gasolene supply, as by dirt in the pipe, an air pocket, or water in the carbureter. Lack of lubrication will stop an engine but not suddenly, without warning.

# Readers with Something to Say

**Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.**

## Trouble with a Barometric Condenser

Some years ago I was employed in a large street-railway power plant when they changed over from surface to barometric condensers. When the new condensers were first installed the rule was to obtain the vacuum before starting the engine as we had done with the surface condensers. One holiday afternoon, one of the engines was held in readiness for service; the engine was oiled up and the vacuum started, but no steam was admitted past the throttle. Holiday loads in street-railway service are rather erratic, hence the engine was not required until about two hours after getting the vacuum.

The assistant engineer finally received orders to start, which he did very carefully. The engine had made only about 25 or 30 revolutions when something gave away in the low-pressure cylinder. We did not know at first just what had given way but upon examination found that the low-pressure piston had been pulled from the rod, having received a slug of water at the crank end.

Salt water was used for condensing purposes and as salt water was found in the cylinder it was at once obvious that the damage was caused by the condenser instead of condensation of the steam.

Shortly after this incident I was transferred to another station of the same company where a new vertical cross-compound engine had just been installed. The exhaust pipe from this engine led to the basement, where a feed-water heater was placed in the exhaust line. From the heater the pipe ran to the top of the power-house roof, where the connections were made to the condenser and to the atmospheric exhaust pipe.

Soon after the engine was placed in regular service, we noticed that at times, when opening the throttle with the vacuum already on the condenser, a loud pounding and surging would occur in the exhaust pipe; in fact, it smashed a 40-inch exhaust elbow located in the basement. Here, again, there was salt water in the exhaust pipe, which was evidence of trouble with the condenser.

After this we received orders to start the engines before building up the vacuum and no further trouble was experienced.

My explanation of the trouble is as follows: When starting the condenser the pipe from the condensing cone to hotwell is probably full of water. When admitting steam to the high-pressure cylinder, or even opening the bypass to the low-pressure cylinder, a considerable quantity of air is liable to be forced from the exhaust pipe into the condenser cone and down to the hotwell. The column of water would then be mixed with air, making it much lighter in weight. The small amount of steam necessary for turning over the engine may, by the time it reaches the condenser cone, be condensed, and represent a tenuity greater than the column of water and air is able to overcome.

A. S. SMITH.

Boston, Mass.

## Results with a Homemade Planimeter

The accompanying diagram was taken from a 12x30-inch Corliss engine running at 90 revolutions per minute with a steam pressure of 95 pounds gage. The scale of the indicator spring was 60 pounds to the inch.

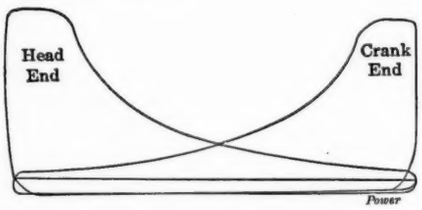



DIAGRAM USED IN COMPARING PLANIMETERS

The areas were found by three methods; first, by using an Amsler polar planimeter; second, by a method of approximation by dividing the card into a number of rectangles; third, by using a homemade planimeter, similar to the one described in the July 12 issue of POWER. The three methods were used for the purpose of comparison and to show the practical value of the homemade planimeter in figuring the indicated horsepower of an engine. The results were as follows:

Total indicated horsepower were 65.65, with the Amsler polar planimeter; 66.3, with the approximate method; 64, with the homemade planimeter.

Assuming that the value found with the Amsler planimeter for the indicated horsepower is correct, the errors involved by using the other two methods were found in the following manner:

$(66.3 - 65.65) \div 65.65 = 0.99$ per cent. Error involved by use of the approximate method,

$(66.65 - 64) \div 65.65 = 2.5$ per cent. error involved by using the homemade planimeter.

J. E. ROCK.

Lansing, Mich.

| Method of Finding Area of Card. | Area Head End, Square Inches. | Area Crank End, Square Inches. | Length of Card, Inches. | Head End M.E.P., Pounds per Square Inch. | Head End I.H.P. | Crank End M.E.P., Pounds per Square Inch. | Crank End, I.H.P. |
|---|---|---|---|---|---|---|---|
| Amsler Polar Planimeter. | 2.85 | 2.72 | 3.937 | 43.5 | 33.6 | 41.5 | 32.05 |
| Approximate Method . . . | 2.864 | 2.735 | 3.937 | 43.9 | 33.9 | 41.8 | 32.4 |
| Homemade Planimeter. . | 2.8 | 2.6 | 3.937 | 42.8 | 33.2 | 39.8 | 30.8 |

## Clogged Radiator

In a large building heated by a single-pipe steam-heating system it was the practice each spring to remove a large radiator in the main corridor and store it until the following autumn. For several seasons the weather had been so mild it had not been needed but on a cold day during the past winter a call came for the radiator and it was hurriedly installed. When turned on it began filling with water and pounded vigorously. The valve was examined and found to be in good condition. Several kinds of air vents were tried and a special drip from the heel of the riser to the radiator was installed. The trouble still continued and, furthermore, an adjoining radiator began to act in a similar manner. The radiator had been turned on end before installing to remove any dirt which might have found its way into it.

It was disconnected for internal inspection and with the water that flowed out there were a large number of walnuts and an assortment of materials used by squirrels for building nests. The nuts had acted as a check valve to prevent the condensation from flowing out, and several which found their way into the valve on the branch line had acted in a similar manner toward the other radiator.

LEWIS C. REYNOLDS.

Willard, N. Y.

## License Laws

Engineers in Massachusetts feel a certain amount of pride in their license laws, yet no one knows better than they that there is room for improvement. In looking over the provisions of the proposed New York State license law, there are found some features that seem to be an advance in the right direction. One is the limitation of special licenses to thirty days; this is an excellent idea. The special license has always been a weak point in the Massachusetts law. No matter how many men holding regular licenses are out of work, a manufacturer may pick up a cheap man and help him to get a license, which license may be renewed without limit. The examiner is supposed to see to it that the employment of the man is a safe proposition, but sometimes influence is brought to bear and the examination is very easy. There is no basis for an argument for the special-license provision on the ground of safety; it merely suits a number of manufacturers who are looking for cheap men, and engineers who are too lazy to do the work required for obtaining a regular license. Some restrictions have been put upon this abuse, but Massachusetts should copy the New York clause on special licenses.

On the other hand, there are features in the New York bill that do not seem good. License laws are in the interest of public safety. If they were for the purpose of raising engineers' wages, there might be some excuse for taxing the engineers to maintain the expenses of the examiners' department; but this is not the purpose, and I doubt if it has any such effect. The public, which is protected, should pay the bills. In this State a fee of $1 is charged for examination. Once a license is secured, there is no further taxation and all renewals are free. New York proposes to tax the engineer $3 for every examination and $2 a year for renewals.

Two dollars from every engineer in the State ought to produce a large annual income, which would make the examining department a paying proposition. But safety, not profits, should be its aim. The engineer, who rises soon after the boss goes to bed, and goes over to the factory to warm it on a winter's morning; who is the last to leave the factory at night; who must be posted on all things in order to get his license; and who for all this receives but a little more than half that which a printer receives for eight hour's work; he is to be made the "goat" once more, and must stand for not only the price of the operation of the new safety measure, but just a little more. Engineers in other States have meekly submitted to such treatment. Here is a chance for New York engineers to protest.

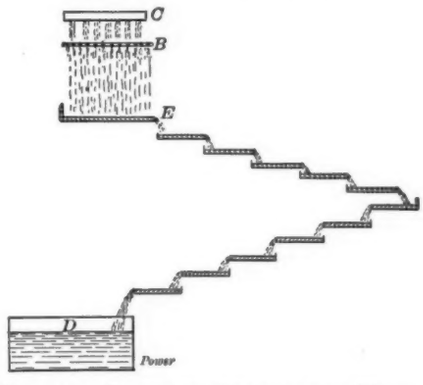
The provision that a man must have had three years' experience as fireman before he can apply for a fireman's license, looks like trying to lift one's self with his own boot straps. How will it work out? Is it necessary in order to insure safety? If a man cannot learn enough around a boiler plant in less than three years to make his employment as a fireman a safe proposition, he must be pretty stupid. Some men learn very quickly. We want these bright men. But such a law will tend to drive them elsewhere. It would seem that the Massachusetts plan of leaving the amount of necessary experience to the judgement of the examiner is more just and more practical.

WILLIAM E. DIXON.

Hudson, Mass.

## A Simple Cooling Tower

The illustration shows a simple form of cooling tower that is adapted to small plants. It can be built within a limited space and at a small cost. The large radiating surfaces exposed to the free circulation of air makes it exceedingly efficient. It is constructed of planks but, unlike many types of cooling towers, it is built in steps. The current of water is first broken up by falling through holes in the bottom of basin *C* and through the wire gauze *B*. It then falls 6 or 8 feet to the platform *E* and gradually finds its way over the various steps to the hotwell *D*. The steps must be true and level and should be placed far enough apart to allow a free circulation of air past the falling water. The hotwell should be large and shallow so as to allow sufficient radiating surface; a groove will answer this purpose very well.




SERIES OF STEPS FOR COOLING THE WATER

In any cooling tower the main object is to bring the water in contact with the atmosphere in particles as small as possible. Many modern cooling towers are now using spray heads through which the water is forced under pressure; but this requires considerable power, and where there is room a simple tower of the type herein described will be found advantageous.

C. R. MCGAHEY.

Sheffield, Ala.

## Repairing a Main Bearing

We had a modern Corliss engine direct connected to a 250-kilowatt, three-phase generator on which the main bearing had twice been allowed to get so hot that the babbitt had started to flow. Each time the quarter boxes had been removed and dressed but the bottom bearing, owing to the difficulty of removal, had not been touched. As a result the bearing always ran so hot that it required continual attention to keep it from burning, hence it was decided that the bottom bearing must come out.

The local machinist claimed that such an operation would take several days and would cost about $50. The conditions of the service were such that this machine could be shut down only on a Sunday; therefore I determined to attempt to accomplish the task during this limited time with the aid of the chief engineer and two assistants.

The armature was slid along so as to clear the field and the cap was loosened on the outer bearing. After removing the cap, the wedge which is used for adjusting the hight of the bearing, was pushed in to its limit and blocking placed under the crank disk. Then the trouble began. The adjusting wedge was designed to be pushed in by means of a large screw, provided with a head for drawing the wedge back again; but when we attempted to draw the wedge back it was found that the head had been broken off and there was no room to get anything behind the wedge to drive it back. This had us puzzled for some time, but we finally conceived the idea of placing the engine on one center, loosening the quarter blocks to give about ½-inch play after putting in a strip of wood to take the shock, and then working the shaft back and forth by admitting steam to alternate ends of the cylinder. This loosened the wedge sufficiently to enable us to push it all the way back.

Next, iron plates were placed between the wedge and the bearing to make up for the amount we had raised the shaft and the wedge was again screwed in and blocking placed under the disk. We repeated the operation until the shaft was raised sufficiently to permit the bottom block to be turned around to the top of the shaft and taken out. The babbitt was found to be quite smooth but all the oil grooves had been completely filled with the metal. It was a short job to cut new oil grooves and put the blocks back into position. The shaft was then lowered by reversing the process by which it had been raised.

Everything was in place and the engine ready to run by 9 o'clock that same evening and the total expense amounted only to the wages of the two extra men for the day.

G. E. MILES.

Salida, Colo.

## Should One or Two Boilers Be Installed?

At the present time there is being built an addition to one of our school buildings. The architect's plans call for the heating to be done by one large boiler. There developed some opposition to this on the part of certain members of the board of education; they believed that two smaller boilers would be better as one could be cut out in mild weather and in the spring and fall, thereby effecting a considerable saving in fuel. I agreed with their views but we were overruled by those in favor of one boiler. I would be pleased to have the opinions of POWER readers as to which is the better plan.

O. LINDBLADE.

Sioux City, Ia.

## Useful Belt Drives

The accompanying illustrations show arrangements of belt drive that do away with cross belts which are often a nuisance. In Fig. 1 the pulleys *B* and *C* are driven in opposite directions by using the idler *D*. This method of belting is

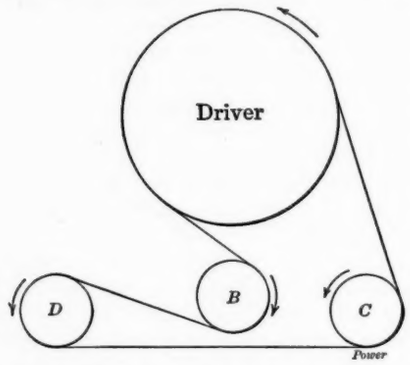



FIG. 1. DRIVEN PULLEYS ROTATING IN OPPOSITE DIRECTIONS

often used to great advantage in operating sawmill carriages, for by using a friction pulley in connection with the

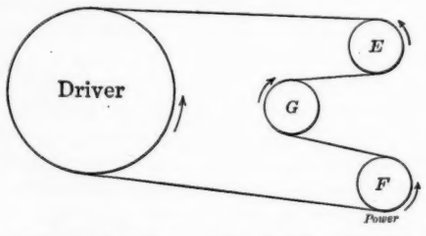



FIG. 2. DRLVER AND DRIVEN PULLEY ROTATING IN OPPOSITE DIRECTIONS

shafts driven by the pulleys *B* and *C* the carriage is driven forward by one shaft and reversed by the other.

Fig. 2 is a variation of this principle. By arranging *E* and *F* to run as idlers and the idler to act as the driven pulley the latter will run in the opposite direction to that of the driver. In both drives herein described the belt is used on both sides.

The foregoing suggestions were embodied in an article by H. N. Saxton, published a few years ago in POWER. They are so simple and useful as to bear repeating.

JAMES E. NOBLE.

Toronto, Can.

## The Care of Pistons

To obtain the best economy from the steam, special attention should be given to the piston. It must be properly lubricated to avoid cutting the cylinder, and must be kept central with the rings and tight to avoid leakage of steam. In fact, it is the part of the engine which should receive the most frequent attention in order to keep it in the best possible condition, and the time spent will yield a better return in dollars and cents than if given to any other part of the engine.

There are nearly as many different styles of pistons as there are of engines. Years ago, when engines were not so highly perfected, the piston consisted of a simple plunger packed with a gasket made of braids of rope yarn tapering for about 18 inches at each end and long enough to go around the piston with the tapers overlapping. The various layers of braids were placed so that the tapers occurred on alternate sides of the piston. Melted tallow was applied with each layer and beaten down with a wooden plug and hammer until the depth of the piston was reached, after which the junk ring was screwed down and more tallow applied. In a later type of piston a single split ring was adopted. The back of this ring was ground or filed to allow for the usual hemp packing behind the ring; the object was for the ring to take up the wear and the hemp and tallow to keep the piston steam tight.

The next improvement was the adoption of a babbitted piston. The piston was made solid and had a deep groove around its circumference. Holes were drilled in the face of the piston connecting with the bottom of the groove. Babbitt was then pounded in till the groove was full; after it was cooled the piston was turned to fit the cylinder tightly. When these pistons became worn the adjustment was made by driving a drift pin alternately in the holes, forcing the babbitt out to the walls of the cylinder. Great care and good judgment had to be used in doing this in order to force it equally at all points, and yet not force out enough babbitt to cause undue friction.

A solid piston having a single split ring in the center was next used, and gradually more rings were introduced and also an expansion ring which assisted in keeping the steam rings tight against the walls of the cylinder. Then the sectional variety was introduced, which consisted of rings in sections, dovetailed into each other and held in place by circular rings or springs. These worked very well, but in all cases it was necessary for the pistons to receive good care in order that they might work well, and the man who points with pride to his engine cylinders and states that they have not been opened for years is not giving his engine the care it should receive.

CHARLES H. TAYLOR.

Bridgeport, Conn.

## Equivalent Evaporation of Steam Boilers

A boiler horsepower is expressed by the evaporation of 30 pounds of feed water per hour at 100 degrees Fahrenheit into steam at 70 pounds per square inch gage. To compare boiler capacities it is usual to reduce both the standard and the given evaporation to an equivalent evaporation of feed water at 212 degrees Fahrenheit into steam at atmospheric pressure (212 degrees Fahrenheit). This is commonly called "equivalent evaporation from and at 212 degrees."

Equivalent evaporation is obtained by

1. Subtracting the heat units in one pound of water at the feed-water temperature from the total heat units in one pound of steam at the given pressure.
2. Dividing the result by 970.4.
3. Multiplying the quotient by the number of pounds of water evaporated under the given conditions.

The product equals the number of pounds of water evaporated from and at 212 degrees Fahrenheit.

Let

$H$ = Heat units in one pound of steam at the given pressure,
$H_1$ = Heat units in one pound of feed water at the given temperature reckoned above 32 degrees Fahrenheit,
$L$ = Latent heat of steam at atmospheric pressure = 970.4,
$W$ = Pounds of water evaporated per hour under given conditions,
$EE$ = Equivalent evaporation in pounds of water per hour from and at 212 degrees Fahrenheit.

Then,

$$EE = \frac{H - H_1}{970.4} \times W$$

An example may serve to illustrate this more clearly.

Given the evaporation of 30 pounds of water at 100 degrees Fahrenheit into steam at 70 pounds gage pressure. Find the equivalent evaporation from and at 212 degrees Fahrenheit.

$EE = (H - H_1)\ W \div 970.4,$
$= [1179.9 - (100 - 32)] \times 30 \div 970.4,$
$= 34$ pounds of water per hour evaporated from and at 212 degrees Fahrenheit.

W. VINCENT TREEBY.

London, Eng.

# Questions Before the House

**Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.**

## Settings of Return Tubular Boilers

The editorial in the issue of August 2 on "Settings of Return Tubular Boilers" calls attention to "a weak point in boiler practice." It is a well known fact that "the side walls and rear connections" often do develop cracks within a short time after the boiler is put into use. It is obvious that there must be a reason for this. In small plants where boilers of this type are largely employed, as in "sugar-houses" on the sugar plantations of Louisiana for instance, the "engineer" in charge thinks nothing of having the boiler settings "done over" at the approach of each grinding season. He has the time to do the repair work, as the boilers are not in use for more than three to five months of the year.

However, it becomes a different thing for the man who has only one boiler, and that boiler has to be kept hot every day of the year. The editorial referred to above recalls to my mind a good example of bad setting. The case I have in mind is that of a large crack running from top to bottom, across the whole side wall, and a little toward the rear, in the brick setting of a small return tubular boiler. The boiler was used to run a machine shop during the day, while the fire was banked at night.

The plant had been in operation only three months, and the whole side wall threatened to come down. This was due, no doubt, to the fact that the setting was built by a "local brick mason," who did not allow for longitudinal expansion. The only way in which to avoid such things is for the makers of return tubular boilers to do as some makers of water-tube boilers do, put in the settings themselves.

J. E. POCHE.

New Orleans, La.

## Packing a Feed Pump

In POWER for August 9, Mr. Kramer asks for information regarding packing for boiler-feed pumps handling water of a temperature ranging from 100 to 195 degrees Fahrenheit. I think that the trouble is not due to the packing or the method of applying it but to the wide range of temperature. I have had the same trouble with a piston-packed pump. My cure, which was effective, was to use one pump when the engine was running and the feed water was at about 200 degrees, and another pump at night and on Sunday when the feed water was practically cold. Before this arrangement was put into effect the packing would last only about five or six weeks. Since that time one packing is good for several months.

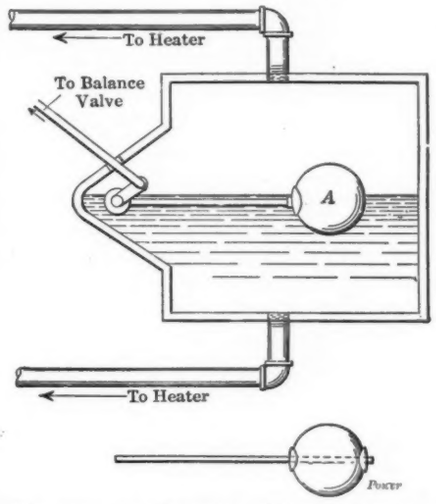
R. L. BEVIS.

Terre Haute, Ind.

## Leaky Floats

In the August 16 issue, Mr. Granfield does not state whether the leaky float has any attachment to it or whether it is independent. Nor does he say whether the leaks are caused by corrosion. His troubles recall to my mind the trouble I experienced some time ago with a float.

The float was used to regulate the water level in an open heater through a series of levers operating a balanced valve which admitted the water fed to the heater. The connection of the stem to the float was as shown in the upper part of the illustration, that is, the end of the stem was threaded and screwed into a collar which in turn was soldered to the float. The leak always occurred at the soldering joint as shown at *A*. No sooner had I repaired the float than the trouble would repeat itself.




ORIGINAL AND IMPROVED CONNECTIONS OF FLOAT AND ROD

Eventually, the idea came to my mind that greater strength would be obtained by passing the stem through the ball and making two joints. This I did, with a stem of sufficient length, as shown in the lower portion of the figure. I never experienced any trouble thereafter.

LUKE MARIER.

Fall River, Mass.

## Central Station versus Isolated Plant

Continuing the discussion of the topic "Central Station versus Isolated Plant," I should like to answer the letter by Warren B. Lewis, in the August 9 issue. He states that electric current is a commodity which is made and sold on much the same basis as any other commodity. I grant that it may be made on the same basis, in other words, by firms in their own building and with their own machinery, but how is it sold? Does not the public allow these companies the right to run their wires through the streets, either underground or overhead as the case may be, continually tearing up the roads to renew their conduit, etc.? In other words, after that current has left the power house it should belong to the public and it does belong to the public. It comes under the head of a Public Service Corporation and under this head it has no right to discriminate. Because I ride up and down town in the street car twenty times a day I am not allowed a lower rate for my fare than the man who may go up and down once a day; because I burn 2000 cubic feet of gas per month, I do not have to pay more than the man who burns 2,000,000 cubic feet a month.

The gas companies of New York City have been forced to make an 80-cent rate by the Public Service Commission. Is there any reason why the electric company should be allowed to go "scott free?" Even I believe that under certain conditions the central-station company should be allowed to make a sliding scale, but these conditions are:

That they charge a certain fixed sum per kilowatt of maximum demand. Take, for instance, the central station located in a busy portion of the city and let it be loaded to its full limit; let it take on some large building, such, for instance, as the City Investing building. It certainly would have to increase its plant to a great degree to supply the current for it, and yet this building would be able to get a cheaper rate, because of its tremendous demand, than the little cigar store on the corner that would not require a single bit of increase in plant in-

stallation. The little fellow has to make up this discrepancy caused by the large sum spent on the increased installation for the big plant. You certainly cannot call this legitimate and fair.

Regarding a cost of three-fourths of a cent per kilowatt-hour this, of course, is ridiculous on the face of it, but with careful operation under the same conditions that the central station has regarding the distance from fuel, etc., even the small isolated plant running anywhere near its full load can certainly make electric current cheaper than it can be bought from the central station, unless the central station is lucky enough to have water power. I know one firm in this city which will guarantee the maximum fuel cost of two-thirds of a cent per kilowatt-hour, under the condition that they are allowed to install the plant and operate it as they wish.

One of the things that I decry greatly regarding the argument of the central station is, that the interest and depreciation on a plant are figured at 10 or even 15 per cent. on the investment, while the Edison company, of New York City, in its report to the Public Service Commission figures on less than 1 per cent. depreciation. This is as much too low as its argument is too high, but it is only another case of central-station exaggeration.

Mr. Lewis is perfectly right in saying that the purchaser has the privilege of going out and employing a consulting engineer. Under certain conditions it is probably more advisable, although perhaps more expensive, to use the central-station current, such as in apartment houses and places of that kind. But he will often run in contact with consulting engineers who are on the quiet employed by the central-station company and who will advise using central-station current whether it is really advisable or not. Let the central station avoid discriminating in rates and it will have as large although perhaps a somewhat different field as it has with its discrimination or, at least, the returns will be as great.

ROSSITER HOLBROOK.

New York City.

## Pumping Layout for Criticism

In looking over the August 9 issue of POWER I noticed H. B. Fillmen's pumping layout which is submitted for criticism. As he wants air-lift information, I will try to help him by describing the system which I am operating. The compressor is of the vertical duplex belt-driven type. Its cylinders are 7x9 inches in size. It was built to run at 190 revolutions per minute and is guaranteed to lift 50 gallons of water per minute with a total lift of 150 feet. We run the compressor at a speed of 140 revolutions per minute and lift water 110 feet. The compressor handles 5000 gallons per hour under these conditions. The discharge pipes in the wells are 2½ inches in diameter; the air pipe is 1 inch in diameter. The small diameter of the air nozzles is about ¼ inch. The submergence is 60 per cent. of the total lift.

Based on the above figures, Mr. Fillmen's outfit would require a compressor having 205 cubic feet per minute displacement and 2½-inch discharge pipes from each well. The pipes should be independent of each other and go over the top of the tank with as few bends as possible.

The submergence should never be less than 60 per cent. of the total lift. The gage pressure is proportional to the weight of the water in the discharge pipe; but as there are alternate layers of air and water, it is hard to guess what the pressure actually is. With a lift of not over 125 feet his compressor would consume about 20 horsepower. My compressor makes a good load for a 7-horsepower motor.

R. E. HENSLEY.

Hico, Texas.

## Design of Rope Drives

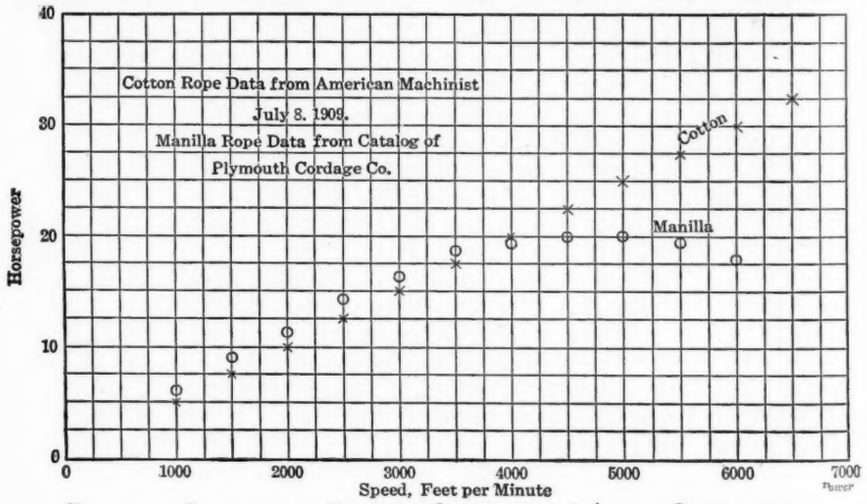
In the issue of August 16, the article by Francis H. Davies, on "The Design of Rope Drives," interested me greatly.

I find that while Mr. Davies refers in the last part of his article to the fact that cotton rope is to be preferred, he has laid out his plot, Fig. 1, with data for manila rope, and limits the speed of a 1¼-inch rope to practically 4500 feet per minute. This, as will be observed by referring to the accompanying figure, does apply to manila rope, but Edward Kenyon quotes very much different results for cotton rope, showing that the effect of centrifugal force is very much less for cotton rope than for manila, so that the power of cotton rope very closely approximates the theoretical value up to speeds of 7000 feet per minute. This is due, no doubt, to the softness and the tendency to cling which exist in cotton rope, and which do not exist in manila rope.

I write this to call Mr. Davies' attention to this fact and to ask him if he has not used manila-rope data in his plot rather than cotton.

HENRY D. JACKSON

Boston, Mass.




EFFECT OF CENTRIFUGAL FORCE ON CAPACITY OF 1¼-INCH COTTON AND MANILA ROPES

## Depreciation

In reading W. H. Bryan's letter on the above subject in POWER for August 9, I noticed some points open to criticism. His remarks on appreciation of real estate seem to be based on the assumption that depreciation will be written against both land and plant. I believe that it is the almost universal practice to exclude the value of the real estate from any depreciation charge; indeed, there are certain firms which write depreciation against the plant less the appreciation on real estate, although most large corporations carry this appreciation as a sort of "secret reserve," unless the difference between book and actual value becomes very marked.

In favor of the fact that the method of charging off a fixed percentage of the previous year's book value never completely wipes out the account, it may be urged that no plant ever becomes entirely valueless, as any machinery, or even old buildings, have a certain scrap value.

There is this objection to a "depreciation fund," in order to make any real difference between this and the method of writing off a fixed amount each year, say 5 per cent. on a 20-year basis (which amount is debited "Operating," and credited "Reserve for Depreciation"), the amount carried in the "depreciation fund" must be put into high-grade bonds, or other similar securities, and these securities held inviolable. This may tie up much-needed funds at very inconvenient times.

A vast deal of the depreciation written

off is what has been termed "depreciation due to advance in art," that is, the early consigning to the scrap heap of good machinery to make way for that which will operate more cheaply. As an instance of the slow depreciation in well kept machinery I remember visiting an iron foundry in 1904 in which a single-acting steam engine was driving a fan blower. The owner told me that to his knowledge the engine had been at work 38 years, and it was old when he first saw it. This shows that what seems to be a popular depreciation period, 20 years, could often be exceeded as far as wear on the machinery is concerned.

DONALD M. LIDDELL.

Grasselli, Ind.

## Effect of Scale

In the brief tests reported in the issue of July 26, a reduction of nearly 20 per cent. in fuel cost is attributed to removing scale from the boiler; but it is clear that if the temperature of combustion had been constant, the difference would have been very much less. With natural draft, clean tubes account for a difference of only 35 degrees Fahrenheit in the flue gases, and with forced draft, only 99 degrees Fahrenheit. If the combustion temperature were 2000 degrees, the fuel saving would be 1¾ and 5 per cent. respectively, due to the removal of scale.

The presence or absence of scale inside the tubes could make no difference to draft or firing efficiency; but both tests before scale removal show bad firing, and both tests after scale removal show careful firing, as shown below.

| | Ashpit Air Pressure. Inches. | Pounds Coal per Square Foot of Grates. | Equivalent Evaporation per Pound Coal. Pounds. |
|---|---|---|---|
| NATURAL DRAFT: | | | |
| Before scaling.... | 0.24 | 11.6 | 6.12 |
| After scaling..... | 0.35 | 15.6 | 7.56 |
| FORCED DRAFT: | | | |
| Before scaling.... | 1.10 | 19.7 | 6.47 |
| After scaling..... | 0.85 | 22.4 | 7.25 |

Note that less draft was used to burn 22.4 pounds of coal than 19.7 pounds. The obvious conclusion is that the correct control of the firing and the draft has a far greater influence on fuel bills than scale can have.

CHARLES ERITH.

London, England.

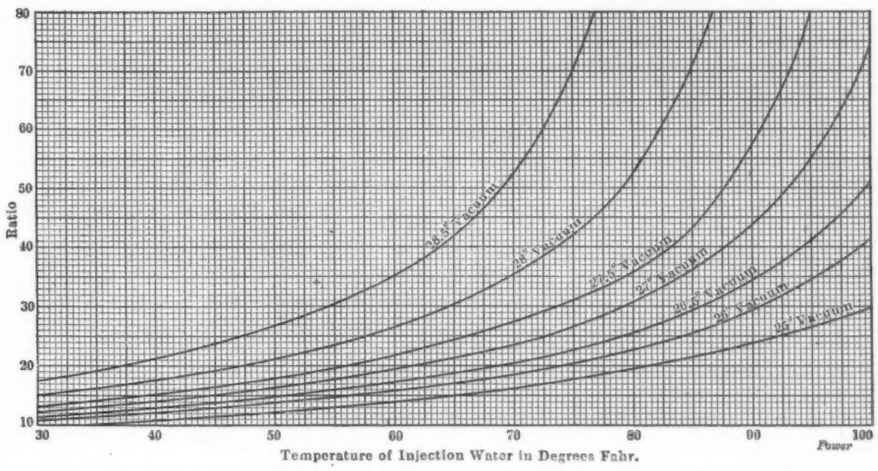
## Condenser Economy

In connection with the article on condenser economy in the August 16 issue, the accompanying curves may be of interest to POWER readers as they show the amount of water required for condensing under various conditions. It is usual to allow a difference of about 5 degrees between the temperature due to the vacuum and that of the mixture, and add this difference to the injection temperature to secure the correct ratio of condensing water. For instance, if a 28-inch vacuum is desired with 70-degree water, the ratio will be about 42 to 1.

While the discharge temperature should be kept as high as possible, the tendency of most engineers is to use excess water to take care of the maximum load. This produces a lower discharge temperature than is desirable. With steady load, however, this temperature can be carried at the maximum.

W. G. STARKWEATHER.

Boston, Mass.




CURVES SHOWING THEORETICAL RATIO OF COOLING WATER REQUIRED AND CONDENSED STEAM

## Pump Problem

I desire to offer a solution to the problem presented in the issue of August 9, by W. E. Thompson, concerning the boiler pressure necessary to operate a 10 and 16 x 8 x 18-inch compound feed pump working against 185 pounds pressure when running at 35 strokes per minute.

In an installation where the pump is set close to the boiler which it supplies and is connected thereto by a well-proportioned steam pipe, the number of strokes per minute depends more on the amount the throttle valve is open than on the boiler pressure. With a long run of uncovered pipe a considerable drop in pressure might take place due to condensation.

As steam is taken full stroke, the secondary pressure bears the same relation to the primary as do the respective cylinder volumes to each other, and we are therefore only concerned with the pressure in the primary cylinder.

It is current practice in designing a feed pump to make the steam-piston area two or three times that of the water piston. The average practice varies between 225 and 233 per cent. This compensates for all losses through condensation, friction, etc.

Referring to the problem, the area of the water piston is 50.26 square inches, and this with the pressure of 185 pounds per square inch makes a total of 9329 pounds to work against. The area of the high-pressure piston is 78.54 square inches. Dividing this figure into 9329 the quotient is 118.5. Then with 118.5 pounds steam pressure the pump would be in equilibrium. Increasing the pressure 25 per cent. gives 148 pounds, which is the minimum pressure required in the high-pressure cylinder to insure satisfactory operation.

The pressure in the secondary cylinder may be determined by the following proportion, neglecting loss by condensation, etc.:

H.P.Cyl. vol. : L.P.Cyl. vol. :: L.P. : H.P.

Hence,

$$100 : 256 :: x : 148$$

$$x = 57.8$$

The pressure, therefore, in the secondary cylinder, with conditions as above mentioned, will be 57.8 pounds.

TRUMANS D. HAYS.

Cambridge, Mass.

## Engineer's Wages

In recent numbers of POWER I have read what some have to say about the engineer and his wages. Possibly to get straight on this important subject we should try to distinguish the *engineer* and the *stopper and starter*. Salary counts for nothing. I have seen good engineers who were working for $45 per month apparently satisfied, while I have run across a lot of stoppers and starters who were receiving $85 per month and who positively could not keep the piston in the middle of the cylinder. The engine would either knock itself hot or the engineer would key it up so tight that it would melt out the bearings and cause a shut down. There is a wider variation among engineers as a class than among the members of any other craft. Lots of fellows will use $45 worth of oil per month; they put it on so freely

that one has to lay boards around the engine foundation so as to be able to get close enough to work on the engine. Other men can be found who will run a large plant on $45 worth of oil per year. This may sound like a fish story, but the facts are drawn from actual cases. In some plants not a spot of oil can be found on the floor, not a spot of rust can be seen and no steam liberated by defective packing can be discovered. The machinery shows a better finish than when it left the shops, due to the engineer's constant industry and pride. This man gets $150 per month. He is an *engineer* not a *stopper and starter.* He can tell when a piston is loose in the pump cylinder. He will not draw up one side of a pump gland so much tighter than the other as to cause the pump to hang.

The wages count for nothing.

We cannot all be the president; but if a man amounts to something and puts his ability on the market in the proper way, he will receive a fair price for it.

The card of a union will not make an engineer. I wager that a union would make the conditions worse than they now are. Let us get right at this matter. If I remember correctly, the N. A. S. E. was formed for a very good purpose. But what has happened? Almost anyone can get into the N. A. S. E. What is the result? Some of those who should wear a button in the lapel of their coat are somewhat ashamed to be a member. There is no use in dodging the truth of this. Let us correct matters. Let a man know something. A school mistress is compelled to have some knowledge before she may teach.

Some men with very little ability get responsible positions. Some one will vouch for the applicant, possibly over the telephone, not knowing anything of the fitness of the man. In he goes. If someone were to ask him to make a Corliss valve work tighter by increasing the compression, he would be lost.

And now, some men are short-sighted enough to say, let us have a "union." Worse than ever, unless all post signs change. This would mean that a good man would have to work for the wages of a poor man, in other words, equal wages. Let us work along some better line. Let us have an examining board which cannot be bought, to see how much a so-called engineer knows. Determine whether he is a two-dollar man or a four-dollar man, then classify him. In many small plants where one man both fires the boiler and looks after the engine, all the valves are bright and clean, the piping is well kept up and no leaks are evident. The engineer often is a $45 man. The job may not be worth any more than this. He is contented, for he has his home. But if he wanted to go out for a better job he could hold it.

Let us stand on our merits not on unionism. Consider some of the N. A. S. E. meetings; lots of good can come from these, but often one gets the floor who knows less than any one else and the posted man goes away in disgust. It would be better to let the best men make the addresses and let the so-called engineers ask the questions "to learn." Do not try to get on top before the foundation is built; you may fall. It is not a matter of what we want to be but what we are. Many so-called engineers working for two dollars per day are not worth it.

I am "from Missouri," when it comes to a union for education I must be shown.

C. R. McGAHEY.

Sheffield, Ala.

## One of Watt's Letters

In the August 16 issue we reproduced a copy of a letter by James Watt to Matthew Boulton. We have received numerous requests to print a "translation," so that with it's aid the text of the letter may be more easily followed.

Our interpretation of the letter is given below. Some of the words, due to the impaired condition of the original manuscript or to cramped penmanship, cannot easily be deciphered. Those of which we are uncertain are set in italics.

"Birmingham, May 10, 1777.

Dear Sir:

Yours of the 8th before me—Salmons man is making wheel engine boiler and seems to do tolerably well.

I am astonished how Joseph deceived himself and us in his *effect* of Bow boiler. I have not seen him since—but I remember he told me that there was a prodigious odds in the quality of some of their coals by others. Did you say whether pushing in the damper to a certain degree did not produce a better or as good an effect as opening the hole in chimney—and was you absolutely certain that no feed got into the boiler during time of experiment?

I think that you are in a good train at Shadwell and recollect no more cautions on that head.

I have made the best apology I could to Mrs. B. though she talks the words of wrath; she is in very good humor and all are well.

When I see the table I can reason upon *copper baylers.*

I saw the Battering Ram, or devil incarnate, go today above 60 strokes per minute and work its own regulators. All the *Hampers* were assembled. They thanked God that Webb could not *make* feeders in. He told them he had one of these and on hand — Moore wants a score of large ones for his own use. The story is all over Birmingham and I expect we shall have customers by the dozen. I imagine that they may be made to work forges and tilts fast enough for any purpose.

Chesburg Iron has turned out damned bad, his piston rod for fuel *bupey* was very well forged by Dixon and upon heating it to float it fell in two at the shooting and Jop rejoiced in his heart thereat, but not Jop nor Joseph could weld it again—nor would it weld to any other iron by no trick.

Dixon is fagotting one out of Sweedish small bars.

The branch of Battering Rams may turn out very *cursed,* as I dare say nobody will attempt them with common engines, and I don't know if they could be made to do. I have invented an admirable thing for opening the regulator which acts by a spring and does it quicker than thought. All Webb's fears are that when a larger one goes at the rate that his does that no body will come near it.

It has demolished all the fixtures many times already, and I suppose must be wholly made of cast iron.

Dangerfield has been here and sett Jop to growling and drinking for two days this week.

I should make myself easy about profits, if any were coming in at all, but a total stagnation as has been hitherto cannot do at all—I am clear that you should make your bargain sure before you leave London, and lett us know what we are to gett with some probably, at least.

As to the Cornish affair. It has also struck me that I should go first and that something should be done before you come.

Adieu.

I wish you a clear head and a firm heart on Tuesday. Pray weigh the coals and observe the quality. Keep well with Rothwell. I think he deserves it.

Yours,

J. WATT."

## Diagrams for Criticism

In answer to J. D. Robertson's inquiry in regard to indicator diagrams in the August 16 issue, if the steam pressure at the throttle is 150 pounds there must be an unreasonable amount of wire drawing in the steam ports. However, I think that the 150 pounds must be at the boiler because with a scale of 60, the pressure at the point of admission is only 95 pounds.

I would advise that a steam-pipe diagram be taken. To do this, tap a hole in the steam pipe above the throttle and place a connection in it for the indicator; hook the cord up to the crosshead in the usual way and take a diagram. If the drop in pressure shows up here then the steam pipe or header is too small; if not, then the fault is in the steam ports, leaky valves or excessive clearance.

WILLIAM H. MAGEE.

Bayonne, N. J.

# Some Points Favoring Compression

By Prof. R. C. Heck

*In the June 28 issue there appeared a review of Professor Dwelshauvers-Dery's experiments with compression, the general conclusions of which were not favorable to compression. The author points out certain limitations in these experiments and proceeds to prove that compression, when properly applied, is advantageous.*

The exposition and review of the experiments of Professor Dwelshauvers-Dery on the effect of compression in the steam engine, published in the June 28 issue, opens up a field for discussion. The experiments were undoubtedly made with care and skill, and the observations may be accepted as correct; yet the conditions were so far from normal as to constitute an extreme, not a typical, case. The investigation was essentially a study of the thermal action upon the cylinder walls; but with a small engine at low speed this action is so exaggerated that deductions from the results are far from being applicable to large engines at usual speeds.

It cannot be denied that much of the traditional theory is very incomplete, failing to give due weight to quite obvious elements of the problem. However, the fact that ridiculous claims have been made for the economy of compression is no reason for going to the other extreme and saying that compression is always harmful, or that it has no effect upon economy. The subject can easily be put into rational form, even though the exact quantitative relations must remain on an empirical and perhaps indeterminate basis.

The simplest way to study the effect of compression is illustrated in Fig. 1. The steam caught in the clearance when the exhaust closes at *E* is compressed by the piston up to the point *F*, and then by the entering steam up to *G*. In expansion this steam is supposed to retrace the curve *GFE*, giving back the work that was expended in compressing it. The only loss of effect is the area *AFG*, which represents a waste of available work through a throttling action upon the entering steam. Some claim that the purpose of compression is to diminish this waste area; others state that the purpose is to fill the clearance space so that less steam will have to flow into it from the boiler before the working stroke of the piston begins.

Consider this latter phase of the subject first. If the expansion curve is carried up to the initial pressure at *J*, the distance *PJ* represents the volume which the total steam in the cylinder would have at that pressure. Of this, *PG* is clearance steam, and *GJ* is fresh working steam, representing the steam received from the boiler. The quotient obtained by dividing the working-area *ABCDEF* by the steam quantity *GJ* is an expression (although incomplete and not in the simplest units) for the thermal efficiency of the engine.

Now consider the other view of the function of compression. If there is no compression, in the ordinary sense, the residual steam in the cylinder at *K* will still be compressed, entirely by the entering steam, up to *L*. The area *EFK* will be added to the work diagram and the amount *LG* to the steam received. Because of the much larger waste area *AKL*, the efficiency according to the ratio of *ABCDK* to *LJ* will be less than it was with compression beginning at *E*. The decrease from *AKL* to *AFG* represents the entire apparent saving by compression; for several reasons the actual saving is considerably less.

A complete analysis of the manner in which the work area *AKL* is wasted would be rather lengthy. Briefly stated, as entering steam flows through a narrow valve, opening into a space of lower pressure, it acquires a high velocity and a part of its stored energy is changed into kinetic energy of the flowing stream. There is no way of applying this kinetic energy usefully, and as the steam comes to rest this energy changes back to heat. When the operation of filling the clearance space is finished, there is in the cylinder a volume *LA* of new steam. If admitted at full pressure behind a moving piston it could have done the work *LAMV*, and then accomplished further work during expansion. In acting upon the clearance steam this admission places only the portion *KLVM* of the available work *ALVM* where it can do work upon the piston. The steam of volume *LA* stands ready for expansion, with the work *ALK*

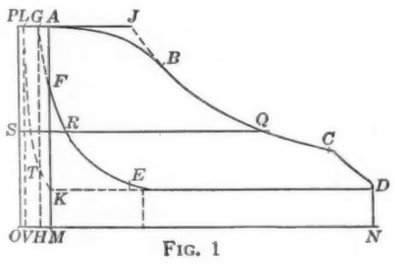


FIG. 1

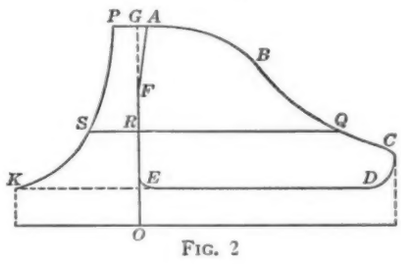


FIG. 2

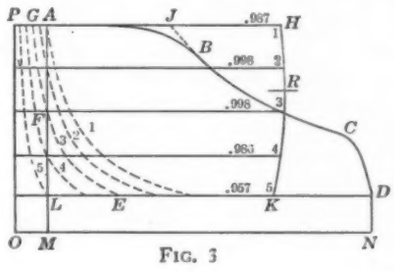


FIG. 3

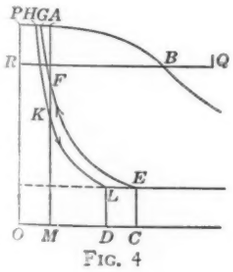


FIG. 4

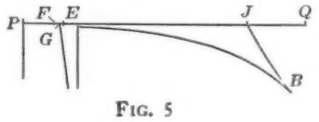


FIG. 5

FIG. 6

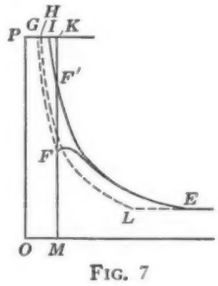


FIG. 7

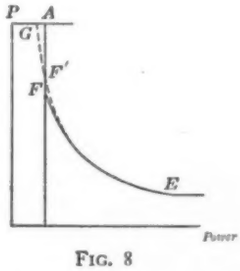


FIG. 8

returned as stored heat; but only a small portion of this heat can be reconverted into useful work. In this simplified description, the effect of the cylinder walls upon the quality of the steam has not been taken into account; this will be discussed further on.

Reverting to the statement that in Fig. 1 the apparent saving by compression is represented by the reduction of the wasted work from *AKL* to *AFG*, the causes which tend to neutralize this gain will now be considered; of these the first is the diminution of the effective expansion. In Fig. 2, through the expedient of rectifying the compression curve *EFG*, Fig. 1 is changed in such a manner as to separate distinctly the expansion operations of the clearance steam from that of the working steam. It may be well to imagine a movable partition or diaphragm separating these two bodies of steam. During the compression *EF*, Fig. 1, this diaphragm is against the piston face; at *F* it begins to act as a dividing wall. In Fig. 2, the volumes are measured from this movable dividing surface; for instance, *RQ* represents working steam and *RS* clearance steam; these are laid off with *R* on the fixed reference line, instead of *S*. Thus the clearance effect is eliminated, and the diagram *ABCDEF* shows the steam action as if it were carried out in a cylinder without clearance.

The purpose of this diagram is to emphasize the fact that, because of the large space which the clearance steam occupies as it expands to low pressure, the effective volume of the cylinder for the expansion of the working steam is reduced. The net result of combining this loss of expansion with the gain due to compression is shown in Fig. 3. Keeping a constant condition of admission, expansion and release, as shown by the curve *ABCD*, the compression is varied from zero to full pressure. For the present, the compression curves are assumed to be alike, all of the form $pv=C$. With each of the five compressions, the effective work area, as *ABCDEF*, is measured and is divided by the steam measure *JG* (*G* being considered movable and representing the end of compression in all cases). Values of the resulting ratios are laid off horizontally from *OP*, on lines through the points of intersection of *AM* with the compression curves; and the curve *HK* shows how this efficiency measure varies with the amount of compression. The maximum point at *R* is found by the purely graphical method of drawing a vertical tangent to the curve *HK*.

From *K* upward there is at first a gain, due to a decrease in the work wasted by the throttling effect of the steam which fills the clearance space. But this is gradually balanced by the loss due to decreased expansion of the working steam till above *R* there is a falling off in efficiency. Evidently, with earlier cutoff and lower release pressure at *C*, this effect of incomplete expansion becomes relatively less, and the point *R* will be higher up the curve. At the same time, with *PJ* becoming shorter, a change in the position of *G* will have a greater relative effect, *HK* will have a greater curvature and the compression will be of more importance as affecting economy. Fig. 3 confirms the generally accepted fact that the action of the ordinary single-valve, variable cutoff gears, which give more compression with earlier cutoff, is just what it ought to be for the large clearance engines on which such gears are used.

The method applied in Fig. 3 is not limited to ideal conditions but is applicable to general use. With actual compression curves, however, the initial point *E* is somewhat more definite than the point *F*. The efficiency curve *HK* can be based either directly on positions of *E*, or on the positions of *F* derived from *E*

Now consider, for simplified conditions, the effect of the initial condensation by the cylinder walls upon the performance of the clearance steam. In Fig. 4 the total steam in the cylinder at cutoff, volume *PB*, would fill the space *PQ* if none were condensed. It is proper to take the clearance steam as dry at *E*; if compressed along the curve $pv=C$, it will be slightly wet at *G*. Assuming that during expansion the steam is homogeneous, the quantity of clearance steam will follow the expansion curve *HKL*, similar to the main expansion curve. At the maximum pressure, the ratio of *PH* to *PG* will be a little less than that of *RB* to *RQ*. Therefore, the clearance steam actually goes through the cycle *EGHL*, with a negative work output, or with a net absorption of the work represented by this closed figure. The line *LE* shows the reëvaporation, which takes place during the exhaust period.

The cycle diagram, as Fig. 4, at once disproves the assumption that the clearance steam gives back during expansion all the work that it received in compression. For present purposes, however, it makes very little difference whether the loss of work due to the cylinder action be divided between the working steam and the clearance steam, or considered as a whole. The important step is the change that must be made in the steam measure by initial volume, the length *GJ*, Fig. 1.

On the upper part of an indicator diagram reproduced in Fig. 5, *BJ* is the expansion curve carried up to admission pressure, *PQ* is the volume the total steam would occupy if dry, *PE* is the dry volume of the clearance steam, and *PF* is the volume of the clearance steam of the same quality as that of the total steam at *J*. The true measure of the steam is now either *EQ* in dry steam, or *FJ* in terms of steam of definite quality. As only relative results are sought in applying the method used in Fig. 3, the latter length will be the more convenient to use.

It has become to be generally accepted that the actual amount of steam condensed by the cylinder walls in the early part of each working stroke is, for a given engine, a nearly constant quantity; that is, it is little affected by a change in the load or by cutoff. If this be true, *FJ*, Fig. 5, for use in the method of Fig. 3, will vary as shown in Fig. 6. The quality of the steam at the beginning of expansion is fixed by the ratio of *PJ* to *PQ*. As the clearance steam increases in amount, so does its absolute shrinkage from *G* to *F*. The result is that *FJ* decreases less rapidly than *GJ* with rising compression, and the upper part of an efficiency curve such as *HK*, Fig. 3, will be inclined a little more toward the left, and the maximum *R* will be somewhat lower.

In the Dwelshauvers-Dery experiments, the engine gave a compression curve similar to *EF*, Fig. 7, and the whole cycle for the clearance steam must have been of the form *EFGHL*. The remarkable thing about these results was the decided increase in cylinder condensation with greater compression. In terms of Fig. 3, all the tests made by Professor Dwelshauvers-Dery were above the maximum point *R*, and the curve *RH* had an extreme curvature toward the left.

As already remarked, this is a very extreme case. The usual result in engines where the conditions tend toward cylinder condensation is a slight droop of the compression curve, such as is shown in Fig. 8. As the steam begins to be condensed as soon as it reaches a temperature above that of the metal, the curve falls below the theoretical curve, rising to *F* instead of to *F'*. The action is of the same character as that in Fig. 7, but does not go nearly so far.

There is still some room for skepticism as to the entire absence of leakage in the experimental engine at Liège. Other experiments, notably those of Callender and Nicholson, quoted in Peabody's "Thermodynamics," have shown that a slide valve, although apparently tight when at rest, may leak considerably when in motion. Assuming, however, that no leakage occurred, the subject may be dealt with as follows : Through compression by the piston, the temperature of the clearance steam is soon raised above that of the inner surface of the cylinder walls. In a small engine, the amount of surface exposed to a given weight of steam is very large; and in a slow running engine there is plenty of time for heat transfer. The result is a

compression curve of the form shown in Fig. 7, and the entering steam comes in contact with walls which are coated with moisture. The chief argument in favor of a steam jacket is not that it makes the cylinder walls many degrees hotter as that it insures their being dry. If, under certain conditions, early compression tends to produce just the opposite effect, we can readily see that it may be decidedly harmful.

Now just as the steam jacket loses its power for good with an increase in the size and speed of the engine, or with a lessening of the influences tending toward initial condensation, so does also this harmful phase of compression rapidly grow less. It is for this reason that the general applicability of the Dwelshauvers-Dery deductions is denied.

The method applied in Fig. 3 can be used equally well in the case of constant power instead of constant cutoff. Then as compression is increased, cutoff will occur later. Again, the upper part of curve *HK* will be bent toward the left, and the maximum efficiency will occur at a lower value for compression.

The proposition which was formerly advanced so frequently, that compression by warming the cylinder walls diminishes initial condensation is now generally discredited. The real purpose of compression is to diminish the waste of available work represented by the area *AKL* in Fig. 1. Referring to the compression curve in Fig. 3 it will be apparent that to raise the point *F* half way from *L* to *A* does far more good than to raise it all the way. Furthermore, every departure from the simple conditions illustrated tends to make the efficiency less with high compression than it appears in that figure. Hence the practical conclusion is "that moderate compression does benefit economy; there is a considerable range, roughly centered on the halfway point (midway between *L* and *A* in Fig. 3) over which change in compression is neutral in its effect; and if compression is carried too high it produces thermodynamic waste.

From the purely mechanical standpoint, the statement that compression helps to bring the reciprocating parts quietly to rest and prevents shock when the motion is reversed is inadequate and not entirely correct. The reversal of the stroke at the dead center involves nothing in the nature of a shock. If the engine were driven by a motor, with the cylinder heads removed, the moving parts would reverse at the end of the stroke without shock; for although the maximum inertia force is present, it varies at that point less rapidly than at any other part of the stroke. The reversal of the pressure on the pin joints would, in this case, occur near the middle of the stroke. Compression does contribute toward quiet running in that it provides for a more gradual reversal of steam pressure which takes place somewhere near the dead center. The rise of pressure shown by a compression curve is less rapid than that caused by the flow of steam past a quickly opened valve into the clearance space, especially if this space is small.

A strong point in favor of compression is that it diminishes the pressure on the pin and shaft bearings while the crank is near dead center. Over this range of motion, the turning moment is so small that the useful work done in turning the shaft is small, regardless of the driving force acting along the connecting rod; a large force, however, produces its full friction effect at the crank pin and shaft journal, because these joints are turning at full speed. By balancing the forward-acting inertia forces of the reciprocating parts near the end of the stroke, compression helps the mechanical efficiency of the engine. It absorbs work which the machine could use only at a disadvantage, and stores it for later delivery when the conditions are more favorable.

# Solution of a Steam Problem

## By C. E. Burgoon

*In the July 19 issue a problem in thermodynamics was cited and solutions requested. The present article furnishes an exhaustive solution of the problem from a purely theoretical point of view.*

Referring to "A Steam Problem," which appeared in the July 19 issue of POWER, the following solution may be of interest to many engineers. The problem as stated was as follows: Given initial pressure 100 pounds absolute; exhaust pressure 15 pounds absolute; quality of entering steam, dry saturated; quality of exhaust steam, 0.9; volume of clearance, 0.4 cubic foot. Find the weight of the steam at *b*, Fig. 1, and its condition, assuming that there is no heat transfer to the cylinder walls.

The volume of the water, which is mixed with the steam, is small compared to the volume of the steam, and has been neglected in the solution. It would not change the results, materially, but would make the solution much more complicated.

The weight of 1 cubic foot of saturated steam at 15 pounds absolute pressure is 0.03805 pound; and the weight of 0.4 cubic foot (clearance volume) equals

$$0.4 \times 0.03805 = 0.01522 \text{ pound.}$$

The exhaust steam has a quality of 90 per cent., hence the weight of steam and water in the clearance volume equals

$$0.01522 \div 0.90 = 0.01691 \text{ pound.}$$

The total heat above 32 degrees Fahrenheit contained in one pound of expanded moist steam is given by the formula

$$H = q + x\,p$$

where,

$H$ = B.t.u. per pound of the steam;
$q$ = Heat of the liquid;
$p$ = Internal latent heat;
$x$ = Quality.

Substituting the constants for steam at 15 pounds absolute pressure and 90 per cent. quality in the foregoing formula:

$$H = 181.3 + 0.90 \times 896.2 = 987.88 \text{ B.t.u.}$$

and the total heat contained in 0.01691 pound of steam equals

$$987.88 \times 0.01691 = 16.70 \text{ B.t.u.}$$

The 0.4 cubic foot of steam, in the clearance volume, must be compressed to 100 pounds absolute pressure, the work (heat) of compression being taken from the admission steam. The volume, quality and heat contents must be determined at 100 pounds absolute pressure in order to determine the amount of admission steam, its quality after doing the work of compression and the quality of the mixture.

Since there is to be no heat interchange between the steam and the cylinder, the compression will be adiabatic; and the entropy at the beginning and end

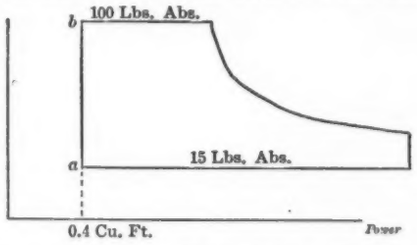



FIG. 1. DIAGRAM PRESENTED IN PROBLEM

of compression will be the same. The entropy at the beginning of compression (15 pounds absolute and 90 per cent. quality) equals the entropy of the liquid, 0.3140, plus 90 per cent. of the entropy of evaporation, 1.4409; that is,

$$E = 0.3140 + 0.90 \times 1.4409 = 1.6107$$

and the entropy of saturated steam at 100 pounds absolute pressure equals the

entropy of the liquid, 0.4748, plus the entropy of vaporization, 1.1273; that is,

$$E_2 = 0.4748 + 1.1273 = 1.6021.$$

It should be noted that the total entropy of saturated steam at 100 pounds absolute pressure is less than the total entropy of the steam at 15 pounds absolute pressure and 90 per cent. quality. This indicates that the steam will be superheated at the end of compression; and the entropy of superheat equals

$$E_s = E_1 - E_2 = 1.6108 - 1.6021 = 0.0087$$

The absolute temperature of the superheated steam, at the end of compression, may be found from the formula,

$$Log.e\,T_1 = (E \div S) + Log.e\,T_2$$

where,

$T_1$ = Absolute temperature of the superheated steam;

$T_2$ = Absolute temperature of the saturated steam;

$S$ = Specific heat of superheated steam;

$E_s$ = Entropy of superheat.

Substituting the value of $T_2$ for 100 pounds absolute pressure, $E_s$ as determined, and $S = 0.55$ in the above formula,

$$Log.e\,T_1 = (0.0087 \div 0.55) + Log.e\,787.4$$

and,

$$T_1 = 800$$

The degrees of superheat $= T_1 - T_s = 800 - 787.4 = 12.6$.

Although steam in a static condition does not contain the external latent heat, the latter must be considered when calculating the work done in compressing the steam. The work done in compression is equivalent to the difference in the heat contents at the beginning and at the end of compression. The total amounts of heat in the steam at the beginning and at the end of compression are given by the following formulas:

$$H_1 = (q_1 + x\,r_1) \times W$$

and

$$H_2 = (q_2 + r_2 + S \times D) \times W$$

where

$H_1$ = Total B.t.u. in one pound of the steam at beginning of compression;

$H_2$ = Total B.t.u. in one pound of the steam at end of compression;

$q_1$ = Heat of the liquid at pressure corresponding to beginning of compression;

$q_2$ = Heat of the liquid at pressure corresponding to end of compression;

$r_1$ = Heat of vaporization at pressure corresponding to beginning of compression;

$r_2$ = Heat of vaporization at pressure corresponding to end of compression;

$x$ = Quality at beginning of compression;

$S$ = Specific heat of superheated steam at that temperature and pressure;

$D$ = Degrees of superheat;

$W$ = Weight of steam in pounds.

Substituting the constants for the steam at the beginning of compression in the first formula,

$$H_1 = (181.3 + 0.90 \times 969.1) \times 0.01691 = 17.81 \text{ B.t.u.}$$

Substituting the constants for the steam at the end of compression in the second formula,

$$H_2 = (298.5 + 887.6 + 0.55 \times 12.6) \times 0.01691 = 20.17 \text{ B.t.u.}$$

The work of compression expressed in heat units equals,

$$20.17 - 17.81 = 2.36 \text{ B.t.u.}$$

The volume of one pound of superheated steam may be found by the formula,

$$P\,V = 0.5962\,T - P\,(1 + 0.0014\,P)\,C$$

where

$V$ = Volume in cubic feet per pound;

$P$ = Absolute pressure in pounds per square inch;

$T$ = Absolute temperature, Fahrenheit;

$C$ = Constant (see Peabody's steam tables, page 76).

Substituting the constants corresponding to the condition of the steam at the end of compression in the foregoing formula may be expressed

$$100\,V = 0.5962 \times 800 - 100\,(1 + 0.0014 \times 100)\,0.211.$$

$V = 4.529$ cubic feet per pound and the volume of 0.01691 pound of steam equals

$$0.01691 \times 4.529 = 0.0766 \text{ cubic feet.}$$

The clearance volume to be filled with admission steam is

$$0.4 - 0.0766 = 0.3235 \text{ cubic feet.}$$

The work required to compress the clearance steam will be taken from the external latent heat of the admission steam. But all of these heat units will not be used for this purpose; and as they cannot be retained by the steam as external latent heat they go to superheat the admission steam; this will be explained later.

The admission steam has raised the clearance steam to 100 pounds absolute pressure; but to determine the quality of the steam, it is necessary to assume a quality, and then check the assumed value. After trying three or four different values for the degrees of superheat, 80 degrees checked fairly close, and the solution will be continued with this value.

The volume of steam at 100 pounds absolute pressure and 80 degrees of superheat as given by the foregoing pressure-volume formulas is as follows:

$$100\,V = 0.5962 \times 867.4 - 100\,(1 + 0.0014 \times 100)\,0.147.$$

$$V = 5 \text{ cubic feet per pound.}$$

The volume of the admission steam in the clearance was found to be 0.3234 cubic foot; and its weight equals

$$0.3234 \div 5 = 0.0647 \text{ pound.}$$

The external heat of one pound of dry saturated steam at 100 pounds absolute pressure is 81.9 B.t.u., and the external latent heat of 0.0647 pound equals

$$81.9 \times 0.0647 = 5.30 \text{ B.t.u.}$$

The heat available for further heating the admission steam equals the total external latent heat minus the heat converted into work to compress the clearance steam, that is,

$$5.30 - 2.36 = 2.94 \text{ B.t.u.}$$

This number of heat units would raise the temperature of one pound of steam at 100 pounds absolute,

$$2.94 \div 0.55 = 5.35$$

degrees (0.55 being the specific heat of steam at this pressure) and 0.0647 pound of steam.

$$5.35 \div 0.0647 = 81$$

degrees. The degrees of superheat were assumed to be 80, and the calculated value, based on this assumption, is 81 degrees, which checks very closely.

The total heat in 0.01691 pound of clearance steam at 100 pounds absolute pressure and 12.6 degrees of superheat equals

$$H = (298.5 + 805.7 + 0.55 \times 12.6)\,0.01691 = 18.80 \text{ B.t.u.;}$$

and the total heat in 0.0647 pound of admission steam at 100 pounds absolute pressure and 80 degrees superheat equals

$$H = (298.5 + 805.7 + 0.55 \times 80)\,0.0647 = 74.29 \text{ B.t.u.}$$

Hence, the total heat in the steam, in the clearance volume, equals the heat in the clearance steam, after it has been compressed to 100 pounds absolute pressure, plus the heat in the admission steam, after it has compressed the clearance steam.

$$74.29 + 18.80 = 93.09 \text{ B.t.u.}$$

and the total weight of the steam in the clearance volume equals the weight of the clearance steam, 0.01691, plus the weight of the admission steam, 0.0647;

$$0.01691 + 0.0647 = 0.08161 \text{ pound.}$$

The heat in 0.08161 pound of expanded saturated steam is,

$$H = (298.5 + 805.7)\,0.08611 = 90.11$$

B.t.u., the heat due to superheat equals

$$93.09 - 90.11 = 2.98 \text{ B.t.u.;}$$

and the degree of superheat equal

$$2.98 \div (0.55 \times 0.08161) = 66 \text{ degrees.}$$

By making use of the pressure-volume formula, the volume of one pound of steam at 100 pounds absolute pressure and 66 degrees of superheat is determined as follows:

$$100\,V = 0.5962 \times 853.4 - 100\,(1 \div 0.0014)\,0.158.$$

$$V = 4.91 \text{ cubic feet;}$$

the volume of 0.01691 pound equals

$$0.01691 \times 4.91 = 0.083 \text{ cubic foot;}$$

the volume of 0.0647 pound equals

$$0.0647 \times 4.91 = 0.317 \text{ cubic foot;}$$

and the volume of 0.08161 pound equals

$$0.08161 \times 4.91 = 0.40 \text{ cubic foot;}$$

Under the conditions given at point $a$ of the diagram, the clearance volume would contain 0.01691 pound of steam at 15 pounds absolute pressure and 90 per

cent. quality and containing 17.81 B.t.u. and under the conditions given at point *b*, the clearance volume would contain 0.08161 pound of steam at 100 pounds absolute pressure and 66 degrees of superheat and contains 93.09 B.t.u.

The constants used in this solution were taken from Peabody's new steam tables, excepting the logarithms, which were taken from "Kent's."

The statement was made earlier in the discussion that "part of the external latent heat of the admission steam would be converted into heat of superheat, and that this would be explained later."

The boiler forces 0.0647 pound of steam out against 100 pounds absolute pressure; and the work done by the boiler equals the external latent heat of the steam or

$$81.9 \times 0.0647 = 5.30 \text{ B.t.u.}$$

Referring to Fig. 2, area *c d h g c* represents the external latent heat (5.30 B.t.u.) of the admission steam; area *c k f h g c* represents the external latent heat (2.36 B.t.u.) of the admission steam required to compress the clearance steam; and area *c d f c* represents the external latent heat (2.94 B.t.u.) of the admission steam which superheats the admission steam.

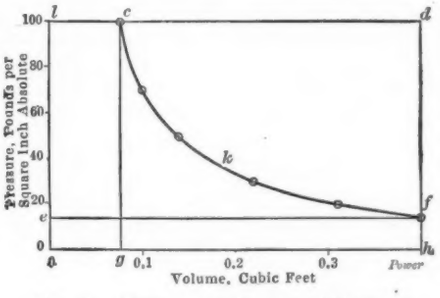



FIG. 2. EXTERNAL LATENT HEAT OF ADMISSION STEAM

Assume a diaphragm to be placed between the clearance steam and the admission steam to prevent steam from mixing or an interchange of heat. Also assume that the admission steam is let into the clearance volume by steps, and that a diaphragm separates each step. The pressure of the steam which fills the first step—say from 15 to 30 pounds—is reduced without doing any work, hence the steam is superheated; when the steam fills the second step—say from 30 to 60 pounds—the steam in the first step is compressed to 60 pounds pressure and is superheated to a higher degree of superheat. The same is true of the succeeding steps until the clearance volume is full of steam at 100 pounds absolute pressure. The diaphragms are now removed, the steam mixes and comes to a constant and common temperature.

The writer invites other engineers to criticize or comment on the foregoing solution; and if it is not right, he sincerely hopes that someone will give the correct solution. The problem contains several points which are not generally understood by technical and practical engineers.

# Minnesota N. A. S. E. State Convention

The attendance and amount of interest shown in this meeting, which was the first in the history of the State organization, was a surprise to many who had the pleasure of being there. In fact, the convention was pronounced as equal to any of the State meetings held in the Central West, and the educational program, consisting of lectures by prominent men skilled in their different lines was more than usually elaborate for a State convention. The character of the mechanical exhibit was also high and all expressed satisfaction as to the results of the meeting.

The convention was called to order in the mayor's reception room, city hall, by H. E. Chase, chairman of the local committee, after which there followed an address of welcome by Alderman Dwyer, representing the mayor. Herbert P. Keller, mayor of St. Paul, also made a pleasing address to the visitors. Governor A. O. Eberhart was represented by his secretary, Ralph W. Wheelock, who commended the activity of the engineers in forming a State organization. Responses to these addresses of welcome were made by F. W. Raven, national secretary, and J. H. Harris, of Chicago. Mr. Raven outlined the history of the organization and brought a message of greeting from all of the national officers.

The educational work included an illustrated lecture by Charles L. Pillsbury, consulting engineer for the State board of control, on the subject of "The Mechanical Equipment of Buildings and Industrial Plants." During the course of the lecture there were shown views of a number of prominent steam plants with their different characteristics and interesting comment was made on the peculiar conditions which in each case determined the method of solving the problems for that particular installation.

J. C. Larson, of Dubuque, Iowa, gave an informal talk on "Steam Turbines," in which he illustrated by sketches and diagrams the difference between the action and reaction type of machines, after which he took up the peculiarities of the different makes of modern equipment, calling attention to the features wherein they differed from each other.

J. L. Barnard talked entertainingly on "Incandescent Lamps," bringing out a number of points which are not generally appreciated by the users.

Another lecture enjoyed by the dele-

DELEGATES TO THE MINNESOTA STATE CONVENTION

gates was by Alex Cunningham, of the Hartford Steam Boiler Inspection and Insurance Company, who spoke on "Inspection of Boilers from the Insurance Standpoint." After giving an outline of the history of boiler insurance he took up the question of boiler inspections and recommended that the hammer test be used exclusively for examination of boilers, as it was his belief that this method, using the eye and the ear, for the detection of defects, is safer and more rational than the hydrostatic test commonly applied. Using the hammer test, an inspector must penetrate to every part of the boiler and will be more liable to detect undesirable conditions existing. Furthermore, in using the hydrostatic test, he believed the boiler is frequently overstrained, sometimes even to the elastic limit of the plates, thereby leaving them in worse condition than they were before the test was applied.

"Analysis of Water Pertaining to Steam" was the subject of a lecture by E. A. Converse, of Chicago. The subject was especially interesting to all present on account of its importance in the daily operation of steam plants. The different reactions in water analysis were shown, and by means of a complete equipment of chemical apparatus the simplicity of the operations was demonstrated. Many analyses of water throughout the Middle West were given and the variation in character of waters in the same district was clearly shown, thus demonstrating the necessity for analyzing the feed water for getting the best results.

One of the most important things to come before the convention was the matter of improving the character of the license issued to steam engineers in the State of Minnesota. In this connection State Senator M. L. Fosseen, in an address on "The License Law, as It Now Is and the License Law as It Should Be." brought out the fact that even the ordinary citizen is rapidly becoming convinced that more rigid examinations and a higher standard of ability should be required from those who are to have charge of steam boilers. Senator Fosseen believed that this question should be agitated without delay in the interest of the public and expressed the hope that some immediate progress in this direction would be made.

Following Mr. Fosseen's address, E. P. Gould, secretary of the Central States Exhibitors' Association, addressed the delegates on the necessity for coöperation between the supplymen and the State association in making a success of their annual meetings.

Entertainment was liberally provided for the visitors, a theater party at the Orpheum being one of the features. An excursion to Lake Minnetonka by trolley and a boat ride on the lake, through the courtesy of the Minneapolis Commercial Club, were thoroughly enjoyed by all.

Officers for the ensuing year were elected as follows: H. M. Germain, No. 7, of St. Paul, president; F. J. Strife, No. 4, of Winona, vice-president; James McGeary, No. 7, of St. Paul, secretary; Albert Johnson, No. 7, of St. Paul, treasurer; P. B. Wells, No. 2, of Minneapolis, conductor; W. E. Strutt, No. 2, of Minneapolis, doorkeeper; Walter Dickinson, No. 2, of Minneapolis, trustee for three years; Joseph Hanson, No. 7, of St. Paul, trustee for two years, and Peter Ulstrum, No. 2, of Minneapolis, trustee for one year; H. E. Chase was indorsed as State deputy.

In closing, St. Paul was chosen as the place of the next meeting.

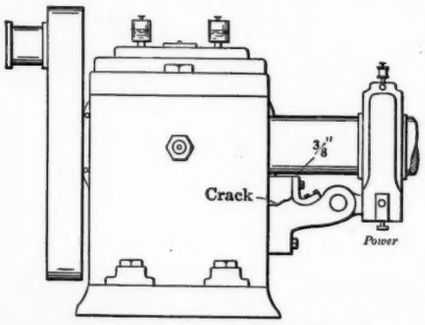
The mechanical exhibit was held in the Grand Army room, city hall, firms having representatives in attendance being as follows:

Williams Brothers Boiler and Manufacturing Company, Minneapolis, Minn.; Bryan-Marsh Company, Chicago; Cadwell & Brown Company, Minneapolis; George B. Carpenter Company, Chicago; Crane Company, Chicago; G. M. Davis Regulator Company, Chicago; Dearborn Drug and Chemical Works, Chicago; Detroit Lubricator Company, Detroit; Electric Machinery Company, Minneapolis; Erie City Iron Works, Erie, Penn.; Fidelity and Casualty Company, New York; Garlock Packing Company, Palmyra, N. Y.; D. M. Gilmore Company, Minneapolis; Fred L. Gray Company, Minneapolis; Greene, Tweed & Co., New York; Hanks-Price Company, Minneapolis; G. M. Harrison, Minneapolis; Hartford Steam Boiler Inspection and Insurance Company, Hartford, Conn.; Hawkeye Boiler Compound Company, Chicago; Hills-McCanna Pump Company, Chicago; Jenkins Brothers, New York; Johnson Service Company, Milwaukee; Keasbey & Mattison Company, Ambler, Penn.; George M. Kenyon, Minneapolis; Keystone Lubricating Company, Philadelphia; King Oil Company, Minneapolis; R. A. and R. M. Laird, Minneapolis; London Guarantee and Accident Company, London, Eng.; Lunkenheimer Company, Cincinnati; Lyons Boiler Works, De Pere, Wis.; William McMillan & Co., Minneapolis; Minneapolis Steel and Machinery Company, Minneapolis; Northwestern Equipment Company, St. Paul, Minn.; W. S. Nott Company, Minneapolis; Osborne High-Pressure Joint and Valve Company, Chicago; Peerless Rubber Company, New York; Penn Oil and Supply Company, Oil City, Penn.; Plant Rubber Company; William Powell Company, Cincinnati; POWER AND THE ENGINEER, New York; Power Equipment Company, New York; Quaker City Rubber Company, Philadelphia; Revere Rubber Company, Boston, Mass.; Sim D. Rollins Company, Minneapolis; *Southern Engineer,* Atlanta, Ga.; Under-Feed Stoker Company of America, Chicago; Viscosity Oil Company, Chicago; R. B. Whitacre Company.

# Temporary Engine Repair Job

Engineers of the South are frequently called upon to do repair work which requires more ingenuity than is exercised by engineers in the North where machine shops are more plentiful. In the case of a releasing gear engine, the piston was broken from some unknown cause, and a piece sticking in the exhaust port caught the valve as it was closing, with the result that the outer bearing of the exhaust valve shaft was broken at a point where it was bolted to the pillow block of the engine, as shown in the illustration.

As the engine was needed, the engineer in charge decided that a reinforcing plate of sufficient strength could be made to fit in the curve of the casting, enabling it to give further service until permanent repairs could be made. A piece of ⅜-inch iron of suitable width was obtained and forged to conform to the shape of the bearing casting, after which holes were drilled in the iron and bearing casting, the latter being tapped for ⅜-inch bolts. When these were secured and the a bolt screwed through the hole in the reinforcing plate corresponding with the hole in the broken end of the casting, the job was found to be so satisfactory that the same bearing is still in use.




DETAIL VIEW SHOWING REPAIR

Open each door to boiler at least once a day and look carefully for leaks at every possible point, noting tube ends, seams and blowoff pipe in furnace. Examine carefully the fire sheets for bulges, bags, or blisters.

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Power and The Engineer

# Editorial

## American Engineering

It has been said, in a mixed spirit of levity and criticism, that Europeans build machines to work, while Americans build machines to sell, and with much humility and mortification we are compelled to admit that some of the examples cited to back up the argument are indefensible. The most recent and striking (no pun intended) of these is the automobile, but as this is not a motor-car paper we shall not go into details beyond the general statement that an enormously greater proportion of American motor cars than of the least meritorious national group of European cars are of bad design and worse construction. But leaving out motor cars, it is uncomfortably true that in general the European design and quality of workmanship embodied in all machinery excepting machine tools and printing presses have been perceptibly—yes, obviously—superior to ours, and Europeans usually account for all of it on the theory that Americans, both builders and users, are in such a hurry that they will not allow sufficient time to be spent on a design to weed out bad features and improve lame ones. That this is true to some extent is undeniable. It is also true that in many cases inferior products have been turned out because the makers were convinced that the "trade demanded" that kind and would not pay the price of first-class manufactures. In still other instances, we are sorry to admit, the builders mean well but do not know enough about engineering to realize that their product is crude; consequently, they have deluded themselves into thinking it excellent.

However, the two kinds of error last cited will eventually be cured, either from the inside or by those who buy, but there exists a more serious condition in American industries than either of these, and one which is a heavy handicap; that is the lack, on the part of the business end, of a proper appreciation of the technical department of a manufacturing establishment. In practically every case where success has been achieved on merit, the men who controlled the business policy were men of either technical training or long practical engineering experience. In almost every case where inferior product is turned out—excepting the two classes first mentioned—the business heads are men whose sole idea is to build just as cheaply as possible and to grind out the machines or apparatus just as rapidly as possible. This class of "manufacturer" is much too prominent in almost all branches of machinery building.

In other words, it is not that American engineering is intrinsically poor but that it hasn't had a full chance because so many of our large establishments are controlled by dollar-grabbers who are either so small-souled that they care nothing for such sentimental considerations as industrial reputation, quality of product, etc., or so narrow-minded that they don't realize that sound engineering is a permanent foundation for which there is no adequate substitute.

Thank heaven there are enough exceptions to prevent us from losing hope, and the number is steadily increasing.

## The Boiler Inspector

As in all pursuits, there are many qualifications required for the making of a good boiler inspector; for example, honesty, temperance and energy; but there are two qualifications of paramount importance, these being confidence in himself and the possession of plenty of backbone. The man who is not certain of himself cannot possibly make a success of boiler inspecting, and unless he has backbone to stick to what he believes is right, regardless of seductive arguments or threats to the contrary, his failure is certain.

The boiler inspector is not only responsible to his company (which generally assumes a considerable monetary risk on his statement of facts), but he is morally responsible for the safety of the lives of those who are required to be in the vicinity of the boilers he inspects.

He should turn a deaf ear to arguments intended to gain his consent for a continuance of conditions that his better judgment tells him are dangerous. In inspecting a boiler plant there are always points noted where improvements might be made; but the inspector must be able to draw a sharp line dividing changes that are necessary for safety and those which may or may not be made at the option of the owner; and such distinction should be plainly evident in his report. Often an owner will make the threat to cancel his policy if certain repairs are demanded, but if the inspector is sure of his ground he is serving the best interests of his company to stand firm, and the chances

are ten to one that he will win out without the loss of the risk. Nothing serves to inspire the boiler owner with confidence in the inspector's ability so readily as having his bluff promptly "called." In the future there will be clear sailing.

The boiler inspector is, of course, only human and therefore will make mistakes. When he is convinced that he is in error a frank acknowledgement will generally serve to place him more firmly in the confidence of the owner. If boiler-insurance companies could only obtain instruments capable of determining the confidence (coupled with ability) and backbone possessed by applicants for positions in their inspecting corps, it would be a simple matter to select the proper candidates.

## Vacuum

The makers of steam turbines claim and realize great advantage from working with a good vacuum; it may be interesting to make a comparison with the conditions for a reciprocating engine.

The reciprocating engine with a large expansion, a pumping engine for example, should have a good vacuum, 28 inches being obtained in good practice. Such an engine may have a reduced mean effective pressure of about twenty-five pounds; a difference of one pound means four per cent. in the economy of the engine, and one pound is the difference between twenty-six and twenty-eight inches.

But the reciprocating engine does not expand down to the back-pressure; good practice for a pumping engine is a terminal pressure of five pounds, absolute. The reason for this is that it takes about two pounds more effective pressure to make the engine go without load; this may be high for a good modern engine, but it is not enough for our present purpose. Now twenty-eight inches of vacuum is about one pound absolute, and if the expansion were carried down to that back pressure, the mean pressure during the expansion would be about three pounds (i.e., the mean of five and one), and subtracting the one pound back pressure would give two pounds mean effective pressure, which would just make the engine go.

Again, the volume of steam at one pound pressure is somewhat more than four times what it is at five pounds; consequently it would require an engine four times as large to expand down to back pressure and that most emphatically would not do.

It must be borne in mind that the reciprocating engine, as previously pointed out, gets a large advantage from a good vacuum without requiring the enormous size required to expand to the corresponding back pressure.

Coming to the case of a steam-turbine, the first idea to get clearly in mind is that the expansion must be carried in the turbine to the vacuum because the steam is discharged at the terminal pressure, whatever that may be, and there is nothing corresponding to the back pressure of the return stroke of a steam engine. But the turbine has two important conditions: in the first place, it is necessary only to add a row or rows of blades to obtain the expansion to a good vacuum, and there is comparatively little disadvantage in having long blades at that end of the turbine; in the second place, the addition of more rows of blades does not add to the mechanical friction of the turbine. The case is more favorable than just stated, for there are certain devices for avoiding undue lengths of blades. In the Parsons turbine they increase the blade angle and reduce the number of blades; this arrangement increases the effective space between the blades and so avoids too great blade length. But increasing the angle also increases the steam velocity, which is even more effective in avoiding too great length of blade. It may be noticed that in marine steam turbines the low-pressure barrel has a large number of rows at the end which have the same length and that these blades are set at a larger angle and especially for the last rows, are spaced wide apart. There is undoubtedly some disadvantage in this arrangement, but at any rate the blades are kept within due bounds.

## Getting Ahead

Many an engineer has wondered why he does not get on with his work and why an acquaintance who is no better informed on engineering subjects, is no more faithful in the performance of his duities and does not keep his plant in any better condition, draws more pay and occupies a better position; also why, when an engineer is required for a new uptodate plant, the acquaintance gets the job. The general answer is that there is some vital difference between the engineer who does the wondering and his friend who gets the job. It is not reasonable to suppose that the other man succeeds in every case by bluffing, because the law of averages would turn the tables sometimes—bluff cannot win in every case. In nearly every case it is merit that enables that "other" engineer to get the job; it is the ability to put to practical use the knowledge he has and that he is constantly obtaining.

Of the thousands of power plants scattered throughout the country, it is improbable that there is a single one that cannot be made more efficient in operation—if the proper things are done; and the man to do the proper things is the engineer. But if he sits in a cushioned chair, sticks his feet up on the desk and puffs away at a cheap cigar, content with himself and the world and with the belief that his plant is better than Smith's, the proper things will not be done.

An engineer does not get an increase in salary for keeping the plant running, for seeing that the oil and waste do not run short, that the engine is wiped off, or that other routine work about the engine room is carried out each day. Not at all. There are hundreds of men of very limited experience who can do all these things, even to holding down the old arm chair.

An engineer cannot go to his employer and demand an increase in salary because he has had the engine started two minutes before the whistle has blown each day for the past ten years. Nor can an increase in pay be expected because the engine has been kept running regularly during working hours for that period. When an engineer can go to headquarters and make intelligent suggestions for improvements of the plant that will save money, and can "make good" after his ideas are carried out, then he is in a position to demand more pay.

What is the advantage of understanding all types of $CO_2$ recorders, for example, if that knowledge is not put to practical use? It will do no good to drop into Jones' place on the way home and advise him that from ten to thirteen per cent. of $CO_2$ should be obtained under ordinary working conditions, when the adviser does not know whether the $CO_2$ in his own plant exceeds five per cent. or not. Nor does it help matters to possess a knowledge of flue-gas analysis and keep it to yourself. The man in the office doubtless knows little, if anything, about this opportunity for saving. It is up to the engineer not only to know about it but to see that his employer knows about it.

The foregoing is only one of many possible opportunities for improvement. The engine cylinder may need reboring, but the president of the company is not usually in a position to know about it, and even when he is it does not tend to advance the engineer's interests to sit down and wait for the head of the establishment to suggest improvements. It is far better for the engineer to be in position to make such suggestions himself.

Summing all this up in a single sentence: It it not the everyday attention to routine work that makes an engineer so valuable that he "gets ahead" steadily, although that is one of the requisites of success; it is the ability to see and make use of opportunities for protecting the interests of his employer.

Probably the best way to make old junk into good machines is to break it up and send it to the foundry.

Don't be stingy with the waste. A clean engine will last longer and do better work.

# Inquiries of General Interest

**Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.**

## *Allowance for Shrinkage Fits*

How much allowance should be made for a shrinkage fit for a crank pin 4 inches in diameter?

J. D. C.

There is considerable diversity in practice and probably no fixed rule that will fit all cases can be given, as much depends on the quality of the material, length of fit, workmanship, etc. In one shop an allowance of 0.0008 inch is allowed for each inch of pin diameter, which for a 4-inch pin would be 0.0032 inch; in another shop 0.001 inch for each inch of pin diameter plus a constant of 0.003 inch is the regular allowance, which for a 4-inch pin would be 0.007 inch, or more than double the allowance first given. Both shops turn out first-class work and their crank pins never come loose.

## *Leakage Through Metal*

Is there any leakage through the pores of metal pipes or tanks containing air or gas under a pressure of 200 to 250 pounds?

V. R. E.

Not unless the material of which the retaining vessel or pipe is made is defective. Compressed air and carbonic-acid gas tanks have been known to carry a pressure of over 2000 pounds per square inch for weeks without showing a measurable loss in pressure.

## *Horsepower of Falling Water*

What horsepower would be developed by a column of water 24 inches in diameter in falling 25 feet?

H. D. W.

As the velocity of freely falling water increases with the distance through which it falls, the column could have a diameter of 24 inches at one point only of its length. Assuming that it has this diameter at the bottom, and calculating its velocity by the formula

$$V = 8.02 \sqrt{h}$$

in which $v$ = velocity and $h$ = head, and substituting 25 for $h$, the result given below is obtained.

$$V = 8.02 \sqrt{25} = 40.1$$

feet per second.

The cross-sectional area of a column of water 24 inches in diameter is 3.141 square feet, and at a velocity of 40.1 feet per second the flow will be

$$3.141 \times 40.1 = 125.95$$

cubic feet per second. At 72 degrees water weighs 62.3 pounds per cubic foot and the energy of 125.95 cubic feet of water falling 25 feet is

$$62.3 \times 25 \times 125.95 = 195{,}167.125$$

foot-pounds per second, which divided by 550, the foot-pounds per second in a horsepower, gives

$$195{,}167.125 \div 550 = 354.8$$

horsepower as the energy of a 24-inch column of water falling 25 feet.

## *Contents of Circular Taper Tanks*

I have a round tapering water tank which is 12 feet across the bottom, 10 feet at the top and 12 feet deep. How shall I calculate its capacity in cubic feet or gallons?

W. M. L.

In form, the tank is the frustum of a cone, the cubical contents of which is found by adding together the areas of the top and bottom and a mean proportional between them, and multiplying the sum by one-third the depth.

| | | |
|---|---|---|
| Area of Top | 78.54 | square feet |
| Area of Bottom | 113.10 | square feet |
| Mean Proportional | 94.24 | square feet |
| Sum | 285.88 | square feet |

$$12 \div 3 \times 285.88 = 1143.52$$

cubic feet, the contents of the tank.

There are 7.48 gallons in a cubic foot. Therefore the tank contains

$$1143.52 \times 7.48 = 8563.52$$

gallons.

## *Boiler Pressure Below the Water Line*

In a horizontal return-tubular boiler the steam gage shows a pressure of 80 pounds. What is the pressure at any point below the water level?

C. S.

A steam gage shows the pressure of the steam in the boiler at the surface of the water. At any point below the surface the pressure is increased by the weight of the water above it. With a pressure of 80 pounds in the boiler, the pressure would be increased 39/100 of a pound for each foot in depth of the water.

## *Width of Butt Straps*

Why is the outside strap of a butt joint made narrower than the inside one?

F. R. A.

The outside strap on a butt joint is made narrower than the inside one, because the width is not necessary for strength and the closer pitch of the second row of rivets makes tight calking possible.

## *Steam Temperature for Any Pressure*

Is there any rule for finding the temperature of steam from its pressure?

W. H. E.

To find the temperature of steam for any pressure multiply the sixth root of the pressure in inches of mercury by 177.2 and subtract 100.

To get the pressure in inches of mercury multiply the steam pressure by 2.041.

## *Covering Sectional Heating Boilers*

Please give me simple directions for applying asbestos covering to sectional cast-iron house-heating boilers.

A. S.

Sectional boilers for house heating usually have a bead molding running along the edge of each section for the purpose of holding the covering in place. The covering should be made into a thin plaster with water and applied in several coats. The first should not be more than ½ inch in thickness and should be put on as roughly as possible. Each coat should be thoroughly dried before the next is applied. The last or finishing coat should be about ⅛ of an inch thick and mixed as stiff as it can be worked and smoothed with a trowel.

## *Effect of Air Supply*

What is the effect upon combustion if too little air is supplied through the fire? If too much air is supplied?

J. A. S.

If too little air is supplied the fire burns slowly and combustion produces carbon monoxide with only one-third of the heating power of carbon dioxide, which is formed when the supply of air is sufficient. If too much air is supplied it causes a waste of heat as it is necessary to heat the surplus quantity of air, and the oversupply of air has a tendency to cool the gases below the igniting temperature.

## *Water Pressure*

Will the water in a tower 120 feet high and 15 feet in diameter exert the same pressure in the mains as a tank at the same hight with an 8-inch riser?

V. K. S.

The pressure per square inch in the mains will be the same in both cases if the water level is the same, as the pressure depends on the hight of the column and not on its shape.

# New Power House Equipment

**What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.**

## Turbine Mechanical Gearing for Slow Speed Steamships

According to a paper read at the spring meeting of the Institute of Naval Architects, by C. A. Parsons, the helical and double-helical gear, of fine pitch, suited to high speeds of rotation was first introduced by Doctor De Laval, of Stockholm, and has been extensively used in connection with his turbine for many years with entire success, and at a moderate cost of maintenance.

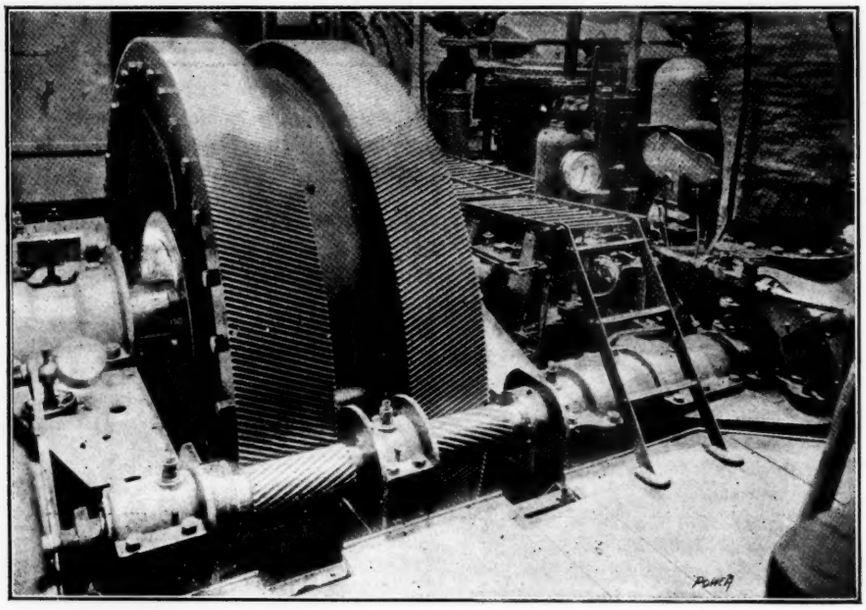


FIG. 1. REDUCING GEAR WITH CASING REMOVED

Mr. Parsons had several experimental sets constructed. One of these was a double-helical gear of the De Laval type, made in 1897, gearing from 9600 revolutions of the turbine to 4800 of the dynamo and transmitting 300 horsepower. The efficiency was estimated by the heat-loss method to be above 98 per cent. This gear was cut in an ordinary universal milling machine without any special precaution as to accuracy and in spite of the obvious irregularity of the teeth it ran well except that it made considerable noise. Gears that have been recently cut by special machinery run with very little noise. A recent experimental set of gearing from 2000 to 400 revolutions, transmitting 300 on a Heenan-Froude water-brake dynamometer, gave a total loss in the gear case, including friction of gear and bearings, of 1½ per cent.

A turbine reducing gear of the design made by the Parsons Marine Steam Turbine Company, Limited, 97 Cedar street, New York City, has recently been placed in the "Vespasian," a slow-speed cargo vessel. The gear is shown in Fig. 1. It consists of pinions on the turbine shaft meshing with large helical gears on the propeller shaft, the teeth in each gear being cut helical. The gears as arranged form a double-helical gear.

In order to eliminate strain on the shaft a flexible coupling is inserted on the shaft between the turbine and small pinion.

The gear is made of cast iron with two shrunk forged-steel rims. The wheel is 8 feet 3½ inches on the pitch circle, having 398 teeth, double helical with a circular pitch of 0.9854 inch. The wheel has a total face width of 24 inches, and the teeth have an inclination of 20 degrees to the axis. The pinion shafts are made of chrome-nickel steel, and have a diameter of 5 inches on the pitch circle, with 20 teeth at 0.9854 circular pitch, the ratio of the gears being 19.9 to 1.

Fig. 2 shows a plan and elevation view of the turbine and reducing gear of the steamer "Vespasian." The weight of the turbine installation was 25 tons less than the reciprocating units which were displaced.

The propelling machinery consists of two turbines in "series," that is, one high pressure and one low pressure, the high-pressure turbine being placed on the starboard side of the vessel and the low-pressure turbine on the port side. At the after end of each of the turbines a driving pinion is connected, with a flexible coupling between the pinion shaft and the turbine, the pinion on each side

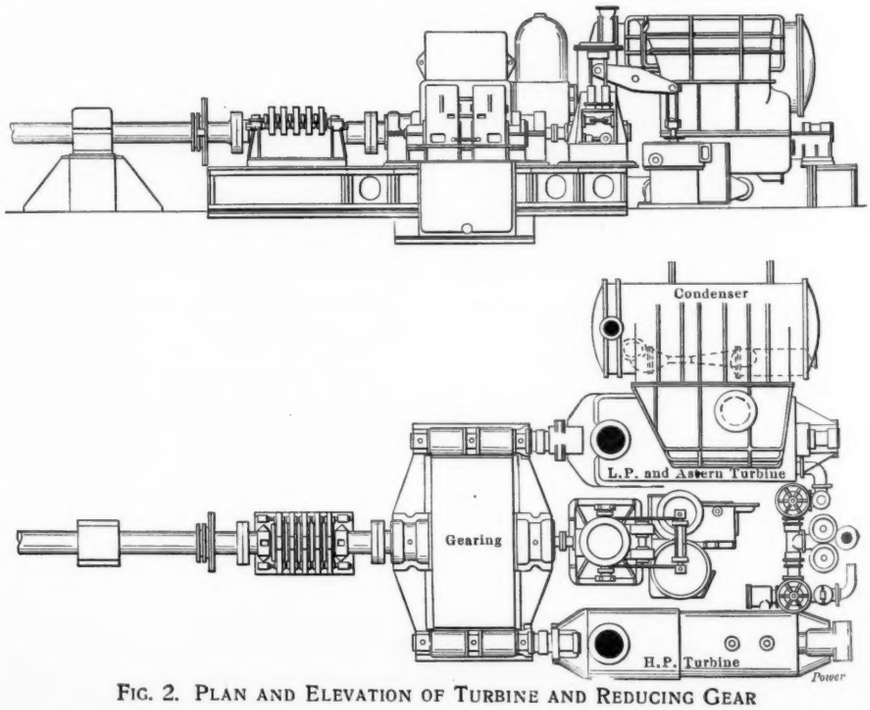



FIG. 2. PLAN AND ELEVATION OF TURBINE AND REDUCING GEAR

of the vessel being geared into a wheel, which is coupled to the propeller shaft.

The following table gives the data and results of a run made off the Tyne on April 11 this year, at varying revolutions:

| | | | | |
|---|---|---|---|---|
| Speed in knots | 8.4 | 9.56 | 10.5 | 10.66 |
| Revolutions per minute | 56.5 | 65 | 71.3 | 73.3 |
| Boiler pressure | 145 | 144 | 140 | 145 |
| Initial pressure high-pressure turbine, lb | 60 | 86 | 110 | 121 |
| Initial pressure low-pressure turbine, in. hg | 15.2 | 12.5 | 7.1 | 5.5 |
| Vacuum, in. hg | 28.8 | 28.8 | 28.7 | 28.5 |
| Barometer, in. hg | 29.9 | | | |
| Shaft horsepower | 456 | 740 | 980 | 1,095 |
| Water consumption per hour, main engines, lb | 9,070 | 12,000 | 14,480 | 15,670 |
| Water consumption, all purposes, lb | 9,670 | 12,620 | 15,120 | 16,370 |
| Water consumption per shaft horsepower, main engines, lb | 19.8 | 16.2 | 14.8 | 14.3 |

A reversing turbine is incorporated in the exhaust casing of the low-pressure turbine. The turbine and pinion-shaft bearings are under forced lubrication, similar to ordinary turbine practice. The teeth of the pinions and of the gear wheel are lubricated by means of a "spray" pipe extending the full width of the face of the wheel. Independent oil pumps are fitted for supplying oil to the bearings and gear wheel.

The high-pressure turbine is 3 feet maximum diameter by 13 feet over-all length, and the low pressure 3 feet 10 inches diameter by 12 feet 6 inches length. The turbines are similar in design to a land turbine, being balanced for steam thrust only; the propeller thrust is taken up by the ordinary thrust block of the horseshoe type, which is fitted aft of the gear wheel.

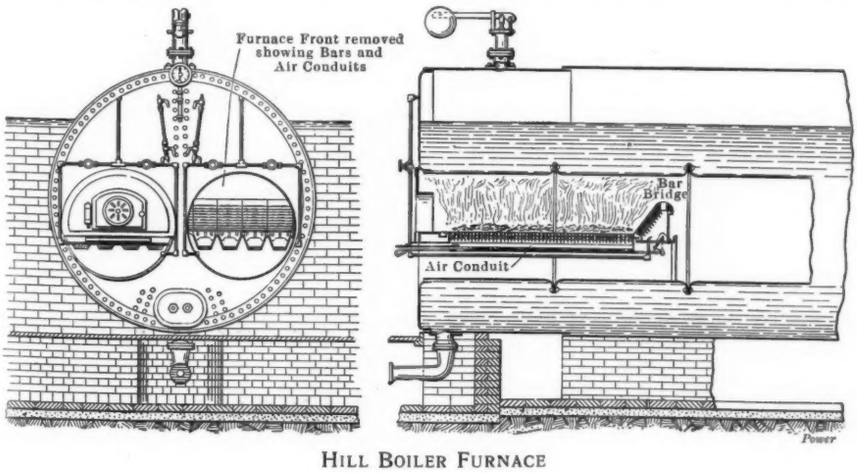
## A New Boiler Furnace

Although a fireman who understands his work can, under general conditions, fire boilers of the Lancashire pattern with economy and without producing an undue amount of smoke, there are cases when the provision of some additional apparatus to effect one or both of these results is beneficial; for instance, when the draft is deficient or when the boiler is too small for its work.

The boiler furnace illustrated herewith is, according to *The Engineer*, being made by Messrs. J. and P. Hill, of Park, Sheffield. The furnace belongs to that class in which the combustion is assisted by means of induced draft produced by a series of small steam jets. The chief features of its construction will be seen in the accompanying drawings. They consist of cast-iron troughs, or conduits, arranged longitudinally along the furnace between the front and back dead plates.

Transversely across these conduits—which are between 8 inches and 9 inches wide, according to the diameter of the flue—short fire bars are placed. The conduits are open at the front end to admit air through the bars, but are closed at the rear end. Their bottom sides are perforated with holes corresponding to other holes in hand-operated slides. These slides permit the stoker to regulate the air supply and riddle the fine ash into the flue. The bars are of special form and are placed much more closely together than is usual. the air spaces being not more than ⅛ inch wide. The bridge is really a continuation of the grate, the bars being specially constructed to fit cast-iron boxes forming the bridge proper, the bridge being finished off at the top by means of a cast-iron cap secured by two prongs that span the box.

This cap has cast in it a large hole through which the mixture of steam and air passes into the stream of gases passing over the bridge. The dead plate is bolted to the furnace front, these being the only bolts used in the erection of the furnace, and the conduits have flanges which fit into recesses in both the front and end dead plates. The boxes forming the bridge are secured together by means of bolts, with which the flames do not come into contact. The draft is induced along the conduits by means of a series of small steam jets, and, the boxes being closed up at the end farthest from the front, the air and steam pass vertically up through the bars, causing the flames to take an almost vertical course before passing over the bridge. The smoke and products of combustion are dealt with by means of a further steam spray through the bridge above mentioned, the amount of air being variable at will by means of the valve shown in the back dead plate. This valve is mounted on spigots, and can be opened or closed at will by means of a handle in front. It will be observed from the sectional views that there are no extraneous pipes or other fittings in the flues to give trouble, and that no drilling of the boiler front is necessary. The whole of the parts can be put in, or removed, in a few hours.




HILL BOILER FURNACE

## NEW PUBLICATION

THE CHEMIST'S POCKET MANUAL (second edition). By Richard K. Meade. The Chemical Publishing Company, Easton, Penn., 1910. Leather, 455 pages, 4x6 inches; 39 illustrations; numerous tables. Price, $3.

Presumably the mathematical tables in this edition were also in the first one (with which the reviewer is not familiar); if so, it is regrettable that some kind friend did not inform the author of their uselessness. Of what practical value is a three-place table of logarithms, or a table of trigonometric functions in one-degree intervals, or a table of roots and powers of numbers ranging from 1 to 100 by units, or—but further enumeration is unnecessary. These useless tables do not detract from the value of the chemical data contained in the book, but they occupy space unprofitably. The tables which are useful are arranged in a sort of haphazard fashion, no definite system of arrangement being evident. Many of the charts in the book are called "tables," some of them are not labeled at all and nearly all of them are poorly executed, as to the mechanical reproduction. Twaddell's name is spelled incorrectly, being disrespectfully written "Twaddle." On page 110 "carbon dioxide" is written for "carbon monoxide," and several similar evidences of carelessness are noticed by the reviewer in various parts of the book.

The manual contains a vast quantity of highly useful material, but the arrangement and treatment of the subject matter are surprisingly amateurish.

## BOOK RECEIVED

"FACTORY ORGANIZATION AND ADMINISTRATION." By Hugo Diemer. McGraw-Hill Book Company, New York. Cloth; 317 pages; 6¼x9⅞ inches; 150 illustrations; indexed. Price, $3.

## Meeting of the A. I. M. E. in the Canal Zone

The members of the American Society of Mechanical Engineers have been invited to join with the American Institute of Mining Engineers during their visit to the canal zone, October 21 to November 15, so far as accommodations will permit. The party sails from New York on October 21 on the Hamburg-American liner "Prinz August Wilhelm," touching at Havana and Kingston *en route* to Colon. A full week will be spent in the canal zone, during which time the party will inspect the work on the canal in addition to holding professional sessions. The cost of the trip, including berth and meals on the steamer, railroad transportation, meals and accommodations at the Hotel Tivoli in the canal zone, will be $200. This rate holds good until September 20, after which an additional charge of $50 will be made by the steamship company. Further details regarding the arrangements for the trip may be obtained from Dr. Joseph Struthers, assistant secretary of the Institute.

# NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

### PRIME MOVERS

WINDMILL. Robin W. Vaughan, Los Angeles, Cal. 968,003.

INTERNAL COMBUSTION ENGINE. Chas. Y. Knight, Oak Park, Ill., assignor of one-half to Lyman Bernard Kilbourne, Chicago, Ill. 968,166.

INTERNAL COMBUSTION ENGINE. Arthur Forbes Scott, Bradford, Eng. 968,200.

TWO-CYCLE GAS ENGINE. Raymond Jacob Leach, Seekonk, Mass. 968,244.

STEAM TURBINE. Jan Proener, Pabianice, Russia. 968,262.

ROTARY ENGINE. Albert C. Savidge, Sunbury, Penn. 968,268.

INTERNAL COMBUSTION ENGINE. Timothy J. Shanahan, Elkhart, Ind., assignor of one-half to Ewing C. Trott, Chicago, Ill. 968,274.

INTERNAL COMBUSTION ENGINE. Thomas A. Martin, Chicago, Ill., assignor, by mesne assignments, to Sears, Roebuck & Co., Chicago, Ill., a Corporation of New York. 968,380.

FLUID-PROPELLED DIRECT-ACTING ENGINE. Wilhelm Mauss, Brakpan, Transvaal. 968,381.

### BOILERS, FURNACES AND GAS PRODUCERS

STEAM BOILER. Alfred A. Olson, Riverside, Ill. 967,961.

FLUID HYDROCARBON BURNER. David P. Moore, Washington, D. C. 968,252.

OIL BURNER AND DRAFT APPLIANCE. Fred A. Stevens, San Francisco, Cal. 968,281.

GAS PURIFIER. Henry I. Lea, Pittsburg, Penn., assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 968,370.

### POWER PLANT AUXILIARIES AND APPLIANCES

IGNITER FOR EXPLOSION ENGINES. Elbridge W. Stevens, Baltimore, Md. 968,280.

WATER-SOFTENING APPARATUS. Kent W. Bartlett, Madison, Wis., assignor to Northern Water Softener Company, a Corporation of Maine. 966,071.

VALVE. Samuel H. Woodbridge, Newton, Mass. 966,150.

WATER-SEALED PRODUCER VALVE. Franklin G. Hobart, Beloit, Wis., assignor to Fairbanks, Morse & Co., Chicago, Ill., a Corporation of Illinois. 967,919.

FURNACE DOOR. Carl J. F. Johnson, Cleveland, Ohio. 967,927.

FEED-WATER HEATER FOR BOILERS. Willard C. Barnes, South Weymouth, Mass. 968,023.

BLOWOFF COCK. Ralph W. Erskine, Pittsburg, Kan. 968,037.

PUMP. William Henry Clark, Hoboken, N. J. 968,126.

OIL BUSHING AND PACKING RING. James J. Coyne and David R. Llewellyn, Shamokin, Penn. 968,132.

CARBURETER. Walter C. Westaway, Rockford, Ill., assignor, by direct and mesne assignments, to J. W. Duntley, Chicago, Ill. 968,215.

PULLEY GUARD. George H. Winslow, South Hadley, Mass. 968,304.

PLUG VALVE. Thomas Keenan, Brooklyn, N. Y., assignor to Keenan Regulator and Steam Fitting Company, New York, N. Y., a Copartnership. 968,362.

PACKING GLAND OR COLLAR. Jules Naeder, Seine-et-Oise, France. 968,390.

CRANK AND YOKE CONNECTION. Harlan P. Wellman, Ashland, Ky. 968,437.

DEVICE FOR GOVERNING AND REGULATING STEAM ENGINES. Hans Christian Holm, Brooklyn, N. Y. 965,904.

HOSE COUPLING. Taylor Evans, Centralia, Wash. 965,998.

EJECTOR. Miles B. Parsons, Birmingham, Ala. 966,048.

FLUE CLEANER. Elza H. Reiter, Elgin, Ill. 966,051.

### ELECTRICAL INVENTIONS AND APPLICATIONS

ELECTRIC FURNACE. Johannese Harden, London, England, assignor to the Gröndal Kjelin Company, Ltd., London, Eng. 967,909.

ELECTRIC FURNACE. James Henry Reid, Newark, N. J. 968,079.

ELECTRICAL POWER TRANSMISSION. John Godfrey Parry Thomas. Chiswick, England. 968,290.

DYNAMO-ELECTRIC MACHINE. Henry G. Reist, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 968,420.

CIRCUIT-CONTROLLER-RETARDING DEVICE. Edward A. Haibleib, Rochester, N. Y. 968,468.

SERIES LAMP RECEPTACLE. Charles D. Gervin, New York, N. Y., assignor to John H. Dale, New York, N. Y. 966,681.

SHORT-CIRCUIT DETECTOR. Frederick C. Reineking, Jersey City, N. J., assignor to Rankand Company, a Corporation of New York. 966,766.

BRUSH HOLDER FOR DYNAMOS. Ernest C. Ketchum, Boston, Mass. 966,839.

ELECTROMAGNETIC SEPARATOR. Frank J. Phillips, Chicago, Ill. 966,855.

ELECTRIC-LAMP SOCKET AND SECURING MEANS THEREFOR. William C. Tregoning, Cleveland, Ohio, assignor to the Tregoning Electric Manufacturing Company, Cleveland, Ohio, a Corporation of Ohio. 966,966.

ELECTRIC WATER HEATER. Herbert N. Roche, San Francisco, Cal., assignor to Thomas B. Gray, San Francisco, Cal. 967,058.

### MISCELLANEOUS POWER PLANT TOOLS

WRENCH. Martin K. Olson, Louisburg, Minn. 967,962.

JACK. John Mosiman, Sedalia, Mo. 968,254.

RATCHET WRENCH. James A. Repass, New Augusta, Ind., assignor of one-fourth to Henry H. Hammer and one-fourth to Ernest M. Hornaday, Indianapolis, Ind. 968,080.

WRENCH. John J. O. Dethlefs and Willis Wisner, Chehalis, Wash. 967,889.

WRENCH. William E. Golden, Fremont, Ohio. 967,899.

WRENCH. Oscar Zimmerman, St. Helen, Ore. 968,106.

JACK. Le Roy Willour, Ashland, Ohio, assignor to the Ashland Manufacturing Company, Ashland, Ohio, a Corporation of Ohio. 968,223.

VISE. Lawrence R. Blackmore, Arlington, N. J. 968,444.

CHAIN TONGS. Philip Farrel, Flushing, N. Y. 968,335.

TOOL FOR SPREADING BOILER TUBES. August P. Gerald, Jersey City, N. J. 968,338.

TONGS. George E. Thurston, Colby, Wash. 967,463.

WRENCH. Lotte Kovatch, Oliveburg, Penn. 967,788.

WRENCH. Malcolm G. Ewer, Detroit, Mich. 967,355.

# ENGINEERING SOCIETIES

### AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

### NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 31 West Thirty-ninth St., New York.

### AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

### AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railroad, Cleveland, O. Next annual meeting at Chicago, Oct. 10-13, 1910.

### WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

### ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

### AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., Dugald C. Jackson; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly.

### AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

### NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September 12-17, 1910.

### UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, John Cope; sec., J. U. Bunce, Hotel Statler, Buffalo, N. Y. Next annual meeting in Philadelphia, Penn., week commencing Monday, August 7, 1911.

### AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

### NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

### OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

### INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

### INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

### NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.